MACHINE DRAWING AND DESIGN
FOR BEGINNERS

HENRY J. SPOONER, C.E.

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AN INTRODUCTORY WORK FOR THE USE OF TECHNICAL STUDENTS

BY

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GENERAL

PREFACE

SOME students taking a regular course of instruction in Machine Construction and Drawing in a day Engineering School or Technical College supplement the instruction given by working from such a book as the Author's "Machine Design, Construction and Drawing." On the other hand, most evening students, during their first year or two, prefer something more portable and less expensive; so, conforming to many requests, and encouraged by the flattering way in which his "Machine Design," has been received in this country and in America, the Author has been induced to prepare this book, in the hope that it may serve as an up-to-date and suitable introduction to the subject, and lead up to his more advanced work.

In arranging the contents of the book, the Author has devoted the first six chapters to the drawing part of the subject, and, guided by his extensive experience, he has treated it in such a way that an intelligent beginner should find it easy to learn the art of making working drawings of simple pieces. The remaining chapters treat more particularly of matters relating to details and machine parts. These are either shown with suitable proportions for various sizes, or are fully dimensioned with the object of making them useful to the young designer or fit for drawing exercises; and occasional suggestions are made as to how such drawings can be best taken in hand. To further assist students and instructors in this direction, suitable drawing exercises are given at the end of most chapters. There are further interesting drawing exercises in Chapter XXIV., suitable for more advanced students, consisting of various engine and machine parts, many of which have appeared in past Examination Papers at the Polytechnic, and a few have been selected from recent papers set by the City Guilds in Mechanical Engineering, and by the Board of Education in Machine Construction and Drawing. The Examination Papers for 1906 and 1908 set by the City Guilds and the Board of Education also appear at the end of the book.

There are also, at the end of most chapters, sketching exercises given, as many of the figures lend themselves to the practice of this indispensable art. Indeed, too much importance can hardly be attached to the cultivation of that clear freehand sketching, having in itself nearly the accuracy of scale drawing, which is such a help to the chief draughtsman in rapidly conveying his ideas. In fact, even a junior draughtsman is expected to sketch with facility. Judged from an art point of view, there can be no doubt that the standard of mechanical draughtsmanship has been considerably lowered since the days of our grandfathers, and the modern practice of first setting out working drawings in pencil, tracing them, and making photographic prints from the tracings for use in the shops, has still further lowered the standard. For in ordinary practice mechanical draughtsmen are no longer called upon to produce drawings with delicate, beautifully joined lines, soft and rich shadows true to geometry, with crisp and dainty surfaces, such as characterized Mr. David Kirkaldy's superb sections of the SS. *Persia*, exhibited in the Royal Academy over half a century ago, and now adorning the office of his famous son, Mr. W. G. Kirkaldy. However, although it rarely happens in ordinary practice

that finished drawings are made, most draughtsmen very properly like to be able to turn out a drawing nicely tinted and finished off with shade lines; but in the ordinary way the student must first master the somewhat difficult art of making a finished pencil drawing, with every line sharp and distinct, and the figuring and lettering bold, neat, and accurate; if this is to be done during an ordinary college course, unless a student has a marked aptitude for artistic work, there is little time available for making pretty or show drawings, and if encouraged to do so systematically it is at the expense of progress along more useful lines. Every student should be encouraged to become proficient in making neat and accurate tracings expeditiously, and up to a certain point the observing ones, whilst acquiring this useful art, will be able to become familiar with many interesting details and with the usual methods of figuring and lettering drawings; but only very exceptional workers could survive a long course of tracing, for it tends to blunt the perceptions and stifle the powers which are required to make good draughtsmen and clever designers. Indeed, tracing has been defined as "a diabolical invention for destroying draughtsmen during the process of their incubation." Now, although mechanical draughtsmen are no longer called upon to produce highly finished drawings, they are expected to be able to transfer to paper any ideas of their chief's or their own, with quickness combined with neatness in such a way that every detail is clearly defined to scale and accurately dimensioned.

It is not usual for writers to refer to such minor details as machine and lever handles, so that young draughtsmen are often left to their own resources to guide them in such matters, therefore a chapter (XIII.) dealing with them has been included.

In recent years much attention has been given to roller and ball bearings, particularly in motor-car work: so the construction of these, and the principles which govern their design, are explained in Chapter XVIII.

In Chapter XIX. the recent and important improvements in spur gearing are described, and although the greatly improved helical gears have made mortise wheels practically obsolete, the consideration that we shall for some years to come still be using some of the old plants led the Author to include mortise wheels in this chapter.

In Chapter XXVI. will be found over one hundred questions relating to the subject, suitable for examinations or for home work purposes.

The Author is indebted to the technical press of England and America for some of the information he has found so useful, and whenever he has drawn from such sources or from technical works, or the Proceedings of scientific and professional societies, he has endeavoured to suitably acknowledge it. In his own training and in writing this book he feels particularly indebted to "Der Konstruckteur," which the genius of Reuleaux gave to the engineering world, and to Professor Unwin's "Machine Design." He has also made references to Professor Goodman's admirable and well-known work, "Mechanics Applied to Engineering."

The best thanks of the Author are also due to the engineers and firms who have kindly permitted him to use their copyright illustrations, or have supplied him with information relating to their specialities. And he cannot refrain from expressing his hearty appreciation of the patient industry of his friend, Mr. E. G. Davey, A.M.I.Mech.E., who made the drawings for most of the illustrations in the book from the Author's rough sketches.

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As the student progresses he will be able to make use of the various figures, whose proportions are given in terms of a unit, for further drawing exercises. In the exercises at the end of most chapters suitable exercises of this kind are suggested.

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DRAWING EXERCISES—*contd.*

TRACING EXERCISES

To assist in selecting figures suitable for tracing purposes, the following particulars are given. The first eight or nine exercises are fairly progressive; the order in which the others are taken may be varied at pleasure. In tracing from the book the corners of the tracing paper may be attached to the page by gum or postage-stamp paper. Set-squares may be used for lines at right angles:—

EXERCISES IN DESIGNING, SKETCHING, AND DRAWING

At the end of most chapters suitable exercises are suggested in the above, and the pages in which these appear are given below—

MACHINE DRAWING AND DESIGN FOR BEGINNERS

CHAPTER I

DRAWING INSTRUMENTS, MATERIALS, ETC.

1. **Hints upon the Selection and Use of Drawing Instruments, Materials, etc.**—The student, having decided to study and practise any kind of mechanical drawing, requires to know what instruments and materials are necessary for the work, and the kind to be obtained, in order to produce satisfactory drawings; but, unfortunately, he often becomes possessed of inferior materials, and a cheap "set" of instruments of foreign manufacture, which are often badly made and for practical purposes almost useless; in trying to use these he handicaps himself considerably at a time when he ought to have for his use the best instruments that can be made.

It is only necessary to visit any drawing-class and examine the work done and the instruments used, to trace the connection between cause and effect in this matter; indeed, if a student who has been studying mechanical drawing for some time fails to make proper progress, it is almost invariably due to the use of faulty instruments.

The student who only wishes to buy those instruments necessary for the making of a pencil drawing, will require a 5″ or 6″ compass with pencil leg (double-jointed and with lengthening bar preferred), a pair of 4″ or 5″ dividers, a bow pencil fitted to hold small circular blackleads,[1] a 12″ rule, preferably a steel one, divided into inches and parts ($\frac{1}{8}$ to 64ths, and 10ths to 100ths), a drawing board, a **T**-square, a pair of set-squares, paper, pins, pencils, and indiarubber. The old-fashioned parallel ruler cannot be relied upon, in fact it is obsolete, but parallel rulers of the *roller form* are very useful for many purposes.

The more expensive instruments are fitted with needle-points, which are preferred by some skilled draughtsmen, but unless they are manipulated with great care and are well fitted with "bolt and nut" arrangement, with shoulders to prevent the points entering too far into the paper, the ordinary conical points are preferable. The latter, in any case, are best for beginners, who lack the light hand necessary for manipulating needle-points properly.

2. **Drawing Board.**—This instrument is used for holding and supporting a sheet of paper flat, whilst a drawing is being made upon it. Care should be exercised in its selection, or trouble may be occasioned by its becoming twisted and out of truth, after very little use. There are many kinds of drawing boards, but the "**Battened**" form is the best, and need only be described. Fig. 1 shows

[1] A few shillings will now buy a small set of very well-made instruments of the English type, with which much useful work can be done. Of course, they must not be compared with the heavier and better instruments turned out by the best English makers, which every student should endeavour to provide himself with, even if he has to buy them separately from time to time.

one of these boards. They should be made of well-seasoned pine, ploughed and tongued together, and grooved half-way through upon the back as shown, being fitted with chamfered battens or ledges of mahogany or oak, to prevent the surface from twisting. The battens are fixed at their centre, to the back of board with screws; and fitted with brass slots let into recesses, and held by cheese-head screws to admit of expansion and contraction of the board with variations of temperature and moisture.

The left-hand edge of the board usually has an ebony strip (which is smoother and harder than the end grain of the soft wood) inserted in it, for the stock of the T-square to slide upon. This strip is sawn through about every inch of its length to admit of expansion and contraction; and projects from $\frac{1}{16}''$ to $\frac{1}{8}''$ beyond the end of the board, which is usually varnished.

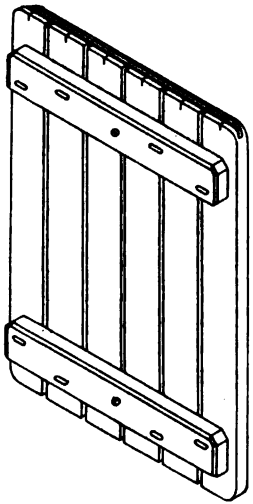


Fig. 1.—Battened drawing board.

The surface of the board may be slightly rounding, viz. convex, from the top to the bottom edge; so that a hollow is not formed under a sheet of paper pinned or stretched upon it. The dimensions of a board most suitable for the exercise work of students is about 24″ × 17″, which takes the half of an "imperial" sheet of paper, or a "medium" sheet. But for drawing office work, a "double elephant" (40″ × 26$\frac{3}{4}$″) is generally used, and for specially large work "antiquarian" (53″ × 31″) is used, these dimensions allowing about 1″ margin between edge of paper and board. Drawing boards for ship work are usually made 100″ × 31″ of 1$\frac{1}{8}$″ pine.

3. Working Position of Board.—The drawing board when in use should be tilted to an angle of about 15°,

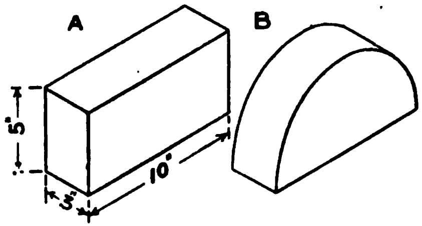


Fig. 2.—Two forms of blocks for tilting board.

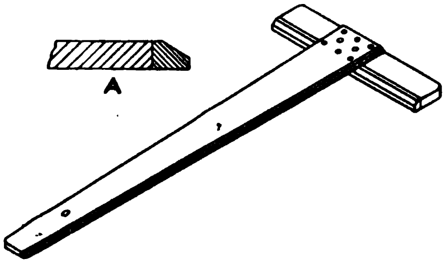


Fig. 3.—English shape T-square.

with the aid of *wooden blocks*, two forms of which are shown at A and B (Fig. 2). Two blocks are used for each board, one being placed under each batten.

4. T-Square.—This instrument is used for drawing long lines perpendicular to an edge of the drawing board; and Fig. 3 shows the "**English shape**," which is best for general purposes. It is made of well-seasoned pearwood, maple, or mahogany. Those of pearwood are the cheapest and answer very well for rough use in a school, but the mahogany ones, with the working edges of ebony, with the pores of the wood stopped with shellac and alcohol and polished (not varnished), are generally used for office work, and should always be used by those who can afford them. An enlarged section of the ruling edge, which should be about $\frac{1}{16}''$ thick, is shown at A on Fig. 3.

5. **To test a T-square in order to see that its Edge is Straight.**—A line should be drawn, using a finely sharpened chisel-pointed pencil (as shown in Fig. 9), holding the pencil quite close to the edge to be tested. Then turn the square over (not end for end), and bring it up to the line, and see if the edge now coincides with it. If so, the edge is straight.

6. **To test whether the Blade is Square with the Stock.**—Draw a line BC, Fig. 4, upon a sheet of paper fastened to the board, parallel to the left-hand edge; at the middle of the line assume a point A, and mark off above and below it AB = AC. Take B and C as centres, and a radius so as to intersect at a distant point C, close to right-hand edge. Join AD. Then place the stock of the T-square against the left-hand edge of the board, and, sliding the square along that edge, see if the edge of the blade corresponds exactly with the line AD. If it does so, the blade is square with the stock.

7. **Set-Squares.**—These are right-angled triangles. They are made of various materials—such as pearwood, mahogany, and other woods, vulcanite, transparent celluloid or pellucid, aluminium, and steel, and are used for drawing short lines perpendicular to, parallel to, or at the angle of the square to one another, in conjunction with a straight edge, T-square, or another set-square.

Two set-squares are generally used, the usual angles and most useful sizes for which are shown in Fig. 5.

Set-squares of pearwood are cheap and useful (if the angles are correct) for students' use, but they are easily soiled, and often

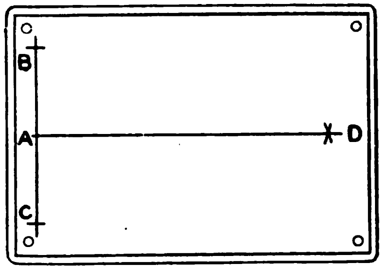


FIG. 4.—Testing T-square.

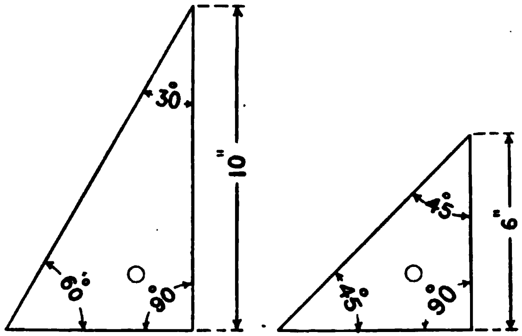


FIG. 5.—The two set-squares.

warp and become untrue. They are not to be compared with those made of transparent celluloid, which on the whole should be preferred.

8. **To test a Set-Square.**—A set-square may be tested to see whether the right angle is correct by placing it against the edge of a T-square, as shown at E, Fig. 6, and drawing a fine line GH against the vertical edge. Without moving the T-square, turn the set edge over, as at F, and bring it up to the line GH to see if it coincides with it. If not, it should be altered until it does.

9. **To test the 60° Angle.**—From a point G on the line AB, Fig. 6, describe a semicircle CD, radius about 12″, and at G, the centre of the semicircle, erect a perpendicular GH. From point C with radius of circle cut the semicircle in J. Join JC, then the angle JCG is 60°, and it can be used for testing the 60° angle.

If J be joined to D, the angle ADJ will be 30°, which can be used for testing the 30° angle of the set-square; but it follows that if the right angle (90°) is correct, and the 60° angle also, the remaining angle must equal 30°, as the three angles of every triangle equal 180°, or two right angles.

Again, if the angles are correctly formed, the long slant edge (hypotenuse) is exactly twice the length of the short edge (base).

The 45° set-square may be tested by bisecting the angle HGD in L and joining LG. Then angle LGD is 45°.

Or, again, if the right angle of the square is correct, and the two edges adjacent to it are of equal length, the angles will be 45°.

10. **Drawing Paper.**—Two kinds of paper are generally used for drawing purposes, viz. "*Cartridge*" *paper* and "*Drawing*"

paper. "**Cartridge**" or "*Machine-made*" drawing paper is used for making details and full-sized working drawings upon. Some of it is sufficiently transparent to be used for tracing paper for large details. It is much cheaper than "drawing paper," and can be obtained either in sheets of various sizes or in rolls up to 62″ wide and 60 yds. long, rendering it extremely useful for diagrams, etc. This continuous paper, as it is called, can also be had mounted on canvas to withstand rough usage. The better quality is of a white colour, and the inferior is of a yellowish tint. The unmounted cartridge paper has two surfaces, a rough and a smooth one; the smooth surface is the proper side to draw upon, and is usually the front side when the water-mark[1] can be read correctly on holding the sheet between the eyes and the light.

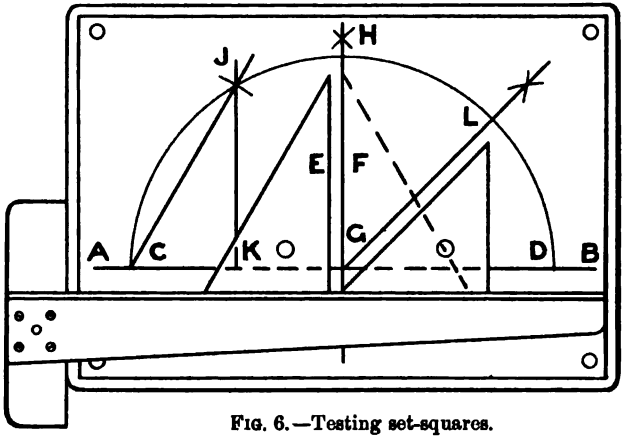


FIG. 6.—Testing set-squares.

Cartridge paper does not usually take tints of colour evenly, but with good paper and care, a very fair effect can be obtained in light tints. But this paper is most suitable for line drawings. It can be obtained in the following sizes, which vary slightly with different makers:—

DIMENSIONS OF DRAWING PAPERS.

	inches.
Demy	20 × 15½
Half-imperial	22 × 15
Medium	22 × 17½
Royal	24 × 19
Super-royal	27 × 19
Elephant	28 × 23
Imperial	30 × 22
Columbier	34½ × 23½
Atlas	34 × 26
Double elephant	40 × 26¾
Antiquarian	53 × 31
Emperor	66 × 47

DIMENSIONS OF CARTRIDGE PAPERS.

	inches.
Copy	20 × 16½
Demy	22½ × 17½
Royal	25 × 20
Cartridge	26 × 21
Elephant	28 × 23
Double crown	30 × 20
Imperial	30 × 22
Double demy	35½ × 22½

11. **Whatman's Hot-pressed Paper.**—For drawings that are to be finished in ink, without colour, the "Hand-made" drawing paper known as Whatman's "Hot-pressed," H.P., "*Smooth*" *or* "*Rolled*" *surface*, is most suitable.

[1] The best qualities only are water-marked.

This paper should also be used for drawings when very fine lines are a necessity, and but little colour is required.[1]

12. Whatman's N.H.P. Paper.—For drawings which are to be coloured or shaded, or are to stand frequent erasing of lines, Whatman's N.H.P. (*not hot-pressed*) or *rough surface* is to be preferred. Its surface will take a fairly fine line, and tints can be laid very evenly upon it. The proper side of the paper to draw upon is that upon which the water-mark of the maker's name can be read correctly when the paper is held between the eyes and the light. This side is generally a little smoother than the opposite one.

13. Quality of "Drawing" Paper.—Drawing paper, *either hot-pressed or not hot-pressed*, is made in various thicknesses, namely, *thin, medium, thick, extra thick*, and *extra extra thick*. The **medium quality** is that generally used for ordinary work. These papers can also be had in continuous lengths, and mounted on union, or white or brown holland; it is then sold in rolls, or by the yard. The two sizes of paper most used in drawing offices are "imperial," 30″ × 22″, or "double elephant," 40″ × 26¾″, but if a smaller sheet is required an "imperial" one is usually halved.

14. Pencils—Different Kinds and Qualities.—The student should only use blacklead pencils of a good quality, such as Stanley's, Faber's, or Hardmuths' *prepared lead*, or Cohen's *Cumberland lead;* inferior makes should never be used for drawing purposes. The following are the requirements of a good pencil for mechanical drawing: It should be moderately hard, of even colour throughout, and durable enough to retain a working point for a long time. It should be easily sharpened, not liable to roll off the board and injure its point, and the lines drawn by it should be easily rubbed out. The ordinary round cedar-covered blacklead pencil, shown at A, Fig. 7, of good quality, is a serviceable pencil, but it easily rolls off the board. To retard the rolling action, some pencils are made hexagonal (Fig. 7, B), whilst Messrs. Stanley & Co. sell a pencil of specially prepared lead, the wooden cover of which is made elliptical, as shown at C, in which this latter defect is removed. The lead of the pencil is rectangular in section and should be fixed in cover, as shown,[2] in order that the wood may properly support the lead, and enable the pencil to be held firmly and the point seen easily when in use. Many draughtsmen use the small solid lead (about $\frac{1}{16}$ of an inch diameter, shown at D, and made by Messrs. Faber, Hardmuth, and others), fitted in a holder, forming what is known as an artist's *ever-pointed* pencil, Fig. 8.

A holder of this kind can be used for years in skilled hands, but the beginner is apt to strip the screw thread when adjusting

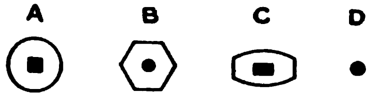


Fig. 7.—Sections of blacklead pencils.

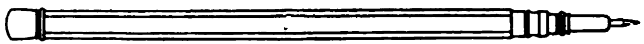


Fig. 8.—Artist's ever-pointed pencil.

the lead, and the instrument becomes worthless, but with proper care the pencil can always be maintained at one length, and be easily sharpened, preferably on a smooth file or piece of glass paper; the lead can then be used up to quite short lengths, which are then available for use in the pencil compasses, thus maintaining an even colour of line throughout the drawing. All the best quality modern instruments are fitted to hold these leads, but many of the older and cheaper forms are fitted with pencil holders of various sizes, and it is difficult to obtain a cedar-covered pencil of sufficient hardness to fit them; these pencils are also more clumsy and difficult to sharpen.

[1] Unstretched paper takes a finer ink line than when the paper is stretched.

[2] Such pencils are occasionally to be seen with the thick part of the lead coinciding with the thin part of the cover.

15. Degrees of Hardness, etc.—Pencils are made in various degrees of hardness, varying from BBBB (the softest) to HHHHHH (the hardest) in wood, and No. 1 to 6 in the solid lead.

Usually No. 1 = BB.	No. 2 = HB.	No. 3 = H.
No. 4 = HH.	No. 5 = HHH.	No. 6 = HHHH.

Nos. 4 and 5 will be found most useful for ordinary work.

16. How to sharpen the Pencil.—For line drawing the pencil should be sharpened to a flat or chisel point, as shown in Fig. 9; this gives a strong point, which retains its sharpness longer than a round one, and it can be worked closer up to the squares, and is more easily sharpened, with the added advantage that the lines are more equal in quality. Needless to say, it is used with its flat side laid against the edge of the T or set square. To make a flat or chisel point to a wood-covered pencil, the wood is first cut away, and the best way to do this is to hold the pencil, as shown in Fig. 10, between the thumb and first finger of the left hand, and to rest it upon the second finger, which should be turned upwards, while the penknife (which should be sharp) is held in the four fingers of the right hand, which should be turned downwards, the thumb of this hand being placed under the pencil to steady it, as shown. A little practice will enable the student to cut a good point with precision and facility, as he has perfect control over the knife, which, should it slip, moves away from the hand. The lead part is best sharpened by

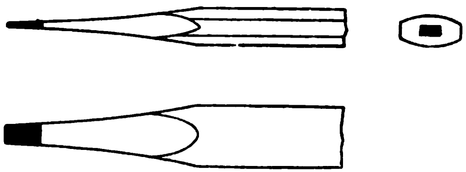


FIG. 9.—Chisel-pointed pencil.

FIG. 10.—Knifing pencil point.

FIG. 11.—Filing pencil point.

rubbing it upon a smooth file, as shown in Fig. 11, after which a stroke or two upon a piece of paper gives it a good finish.

A 6″ smooth hand file, or a 4″ or 5″ triangular saw file, should be preferred. If a file is not available, a piece of fine emery

paper or cloth, "F" or "FF," or glass paper, "O," fastened to a strip of hard wood about 6″ long, 1″ wide, and $\frac{1}{4}''$ thick, is a good substitute, or small blocks, containing about 16 surfaces of glass paper, especially made for pencil sharpening, may easily be obtained. The latter are very useful for giving the pencil a finer point than can be made with a knife alone, and when the surface is worn the damaged thickness can be torn off and a fresh surface exposed for use. However the point may be produced, a few strokes on a piece of blotting or soft paper will give the point a beautiful working edge. Short pieces of pencil—under 3″ in length—should not be used for line drawing unless fitted into a rigid holder, as sufficient command cannot be obtained over them.

17. Compass Pencils.—The points of compass pencils should be made narrower than for straight-line purposes, and must be carefully adjusted so as not to draw a thick line; indeed, the beginner is more likely to do better work with a **conical**-pointed lead in his compasses. It is not enough to start with a good point, *its sharpness must be maintained*, and this requires constant attention. As soon as the lines appear to be too thick, one or two strokes of the pencil upon the sharpener will restore the point.

18. The Conical-pointed Pencil.—For the making of freehand sketches, dimensioning, or descriptive writing upon a pencil drawing, it is desirable to use a softer pencil than that used for line drawing (such as a No. 3 or 4, or H or HB), and to sharpen it to a long conical point, as shown in Fig. 12.

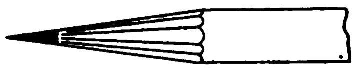


FIG. 12.—Conical-pointed pencil.

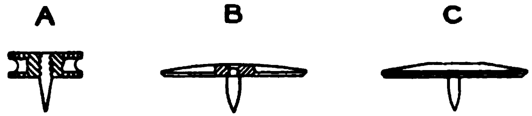


FIG. 13.—Three different forms of drawing pins.

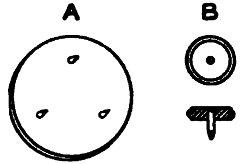


FIG. 14.—Horn and metal centres.

Several pencil sharpeners are sold for this purpose, but they do not produce a good long point, such as draughtsmen pride themselves upon, and which can be readily made as previously explained. *The point should on no account be moistened* when used, as marks made by it in that condition are very difficult to erase.

19. Drawing Pins.—To secure the paper to the drawing board either drawing pins or paper clips are used, or, if the drawing is a very important one, the paper should be **stretched**; particularly is this necessary if the drawing is to be highly finished by shading and colouring.

There are many kinds of drawing pins, three of which are shown in Fig. 13. That at A consists of a brass head, with milled edges, with a steel pin screwed or riveted into it. This form projects too much above the surface of the paper, the height of the head preventing the T-square from lying flat upon the paper, and the edge of the T-square is injured by coming into contact with it. The pin-point is often badly fastened into the head, frequently unscrewing from it, and the shape of the point does not enable it to hold well into the board.

The one shown at B is formed of a disc of brass or electrum, turned circular to the section shown, and has a steel pin; the head is about $\frac{1}{16}''$ thick at the centre, and the upper surface is convex, and thinned towards the edge, which is rounded and milled, so that the thumb-nail can be used for removing it from the board. The pin shown at C is similarly made, only its upper surface is flat and bevelled instead of being rounded.

The two latter forms are those generally used in drawing offices, and they are recommended; the most useful sizes are $\frac{9}{16}''$ to $\frac{3}{4}''$ diameter.

20. Horn and Metal Centres should be used for preventing the centre point of the compasses from making a large hole in the paper whenever a number of circles are described from the same centre. They consist of small discs of transparent *horn* or of electrum, as shown at A and B, Fig. 14.

21. Indiarubber.—The ordinary dark coloured native or bottle indiarubber is best for removing pencil marks when the best blacklead has been used, especially if the drawing is to be coloured, as it does not injure the surface of the paper when carefully used, and if the rubber becomes dirty or sticky with use it can easily be restored by boiling in clean water for a short time. Fine vulcanized grey or white indiarubber is very largely used, especially for blacklead pencil marks upon machine-made paper, the lines of which are hard to erase, but care should be taken that the rubber is soft and not too highly vulcanized or gritty, or the surface of the paper will be injured and rendered unfit for colouring purposes.

22. Measuring Rules.—A 12″ steel rule, divided into inches, with divisions of 8ths, 16ths, 32nds, and 64ths on one edge, and 10ths and 100ths of an inch on the other, will be found useful. The more simply a rule is marked the better for ordinary use, especially when foreign measures are concerned. It is much better to use separate rules than to crowd a number of different divisions on a single one, which often lead to serious errors being made. This steel rule should be plated with nickel, or an alloy of platinum, the former prevents rust, and the latter rust and also discoloration.

Care should be exercised in placing the points of the compasses upon a steel rule in a *normal* direction, or they will be injured. Fig. 15 shows how, by inclining the compasses to the rule, the sides of the points may be made to rest in the cuts or divisions without injuring the points. The figure also shows how the compasses and rule should be held if the right hand is to have complete command over the former in adjusting the points to take off any required dimension. An edge of the rule may also be directly placed on a line and a dimension pricked off by sliding the pricker down the divisions of the rule, but this requires great care. The accuracy of the *steel* rule and its durability make it superior to any other at the command of the draughtsman. As the student and draughtsman are now so frequently called upon to set out work with metric measurements, there is no reason why the back of the steel rule should not be divided into centimetres and millimetres.

FIG. 15.—Application of compasses to rule.

23. Drawing or Ruling Pens.—In most full sets of instruments there are two ruling pens, a large one and a smaller one, called a *fine ruling pen.* The best type of these pens is *jointed*, so that when the screw is taken out one of the nibs can be moved away from the other about its hinge or joint for cleaning purposes. The cheaper pens are made without this joint, and there is a difficulty in cleaning them. This is best done in any case by drawing an edge of a damp duster between the nibs and not by scraping them with a knife or file. All pens must be frequently cleaned when in use, and should never be put away without being completely freed from ink and dirt. They should also be sufficiently unscrewed to prevent the nibs remaining in contact when not in use.

24. Indian Ink.—It is well known that ordinary writing ink is unsuitable for use on drawings, as, although it is more or less indelible, it has not the blackness and body that are considered necessary, to say nothing of the corrosive action of such inks on steel, which alone would preclude its use in the ordinary *drawing pen.* In addition to these objections it runs too freely

from the pen and blurs when touched by a brush in colouring. The only ink that satisfies all the draughtsman's requirements is known as *Indian ink;* this ink, when properly used, produces a clean, dense, jet-black line, and, being free from acid, it does not corrode the instruments: it can be had either in a solid or liquid form.

The quality of Indian ink differs very much, but if good the stick will have a brownish glazed appearance at the end after being used.

25. Liquid Indian Ink.—Some makes of this ink are very good, but the majority are very indifferent. It is often purchased by students because it is cheap in first cost (6*d*. or 1*s*. per bottle) and saves the trouble of mixing. But as a rule it is not so black, is liable to dry up and deteriorate in quality after being opened, and the lines drawn with it lack the beautiful jet-like appearance so characteristic of good Indian ink; it cannot be used for shading purposes with much success, and the bottles are liable to get upset.

26. Colours, etc.—The best water colours only should be used for tinting mechanical drawings; these may be obtained in cakes, hexagonal sticks, or small pans, and as very few will suffice for the student to begin with, it is of no advantage to purchase inferior ones. If cake or stick colours are used, they are ground up with water in a saucer until of the required depth of tint. **Moist water colours in pans** are to be preferred for students' use and for drawing office work. The pans in which the colours are placed are of china or porcelain, covered with a suitable wrapper, which should not be removed, but cut through three sides at the top of the pan with a sharp penknife, to form a lid to protect the colour when not in use.

The most useful colours and the materials they are used to represent are given below. The first four will suffice if only ordinary metals are to be indicated; the others are required when the other materials of construction are to be shown in colours.

Prussian blue	to represent	**wrought iron and dimension lines**
Payne's grey	,,	**cast iron**
Crimson lake	,,	**centre and datum lines**
Gamboge, or Indian yellow	,,	**brass and gunmetal**
Yellow ochre	,,	**stone**
Burnt sienna	,,	**wood**
Sepia	,,	**leather**
Light red	,,	**brickwork**
Indigo lead	,,	**lead**
Burnt umber	,,	**packing**
French ultramarine	,,	**water**
Prussian blue and crimson lake	,,	**steel**

27. Saucers for mixing Colours.—Saucers for mixing colours in are of various kinds; but the most useful ones for students or office use are the cabinet nests of white china, which are sold in sets of five and a cover. They vary from $2\frac{3}{8}''$ to $3\frac{3}{4}''$ diameter, the largest size being most useful. With these saucers, colour left over from a previous wash can be remixed and used up. Dust can always be kept out by piling them up together and putting the cover on.

28. Brushes.—For colouring drawings the student will require at least two brushes, the most suitable being a "**middle swan**" and a "**small goose,**" preferably of red or brown sable hair. He will also require a camel-hair water brush of about "**large swan**" size, for transferring water to the saucers, etc.

CHAPTER II

PRINTING, TRACING, SHADING, ETC.

29. Printing, etc.—The following style of lettering, which should be neatly written with an ordinary writing pen, is most suitable for notes or remarks on a drawing:—

a b c d e f g h i j k l m n o p q r s t u v w x y z

In drawing office practice it is usual to **stencil**[1] headings and titles, etc., in plain letters, such as the following, the size varying from $\frac{1}{8}''$ to $\frac{3}{4}''$, according to the size of the drawing; for example, the heading or title on *medium* or *royal* size sheets would be in good proportion if made with $\frac{3}{8}''$ or $\frac{1}{4}''$ letters, or with $\frac{3}{8}''$ and $\frac{1}{2}''$ letters for *imperial* and *double elephant* respectively; and such titles as *plan, elevation,* etc., with $\frac{1}{8}''$ or $\frac{3}{16}''$ letters:—

ABCDEFGHIJKLMNOPQRSTUVWXYZ
1234567890

Although most of this printing is done by stencilling, students should endeavour by practice to do it neatly by freehand, to enable them to proceed when stencil plates are not available. The quality of printing and writing upon a drawing greatly adds to or detracts from its appearance.

30. Working Drawings of machinery are made in such a way that the form and size of every detail are clearly shown for the guidance of those in the works. The rule is to make them to as large a scale as possible, generally full size for all small details, and $\frac{1}{2}$ and $\frac{1}{4}$ full size for larger ones. Such drawings are first carefully set out in pencil and then inked in, all parts cut by section planes being cross-hatched with sectional lines indicating the materials they are made of, in accordance with the shading shown in Art. 61; or, alternately, they are coloured[2] to indicate the materials, as explained in Art. 26. The edges of **surfaces** that are **to be machined** are usually coloured with a narrow band of a deeper tint. The next step is

[1] A good deal of practice is necessary to enable the beginner to do this neatly. He usually commences by making the stencil brush too wet, which causes the ink to flow between the stencil plate and paper. The best expedient is to recess a piece of Indian ink in a thin block of wood, and, after wetting the brush, rub it over the ink and wood till it is dry enough to use on the plate.

It is best to start from the middle of a title when stencilling, so as to get it quite symmetrical with the drawing. This can easily be done by counting the letters which come each side of the centre, allowing one for each interval between two words.

[2] Drawings from which tracings are to be made for reproduction by photographic printing are, of course, always section-lined and not coloured.

to ink in with red ink[1] the **centre lines**, and the **dimension lines** with Prussian blue.[2] The arrowheads and the dimensions should be now neatly written with an ordinary writing or mapping pen, care being taken to make the dimensions bold and neat, so that they can be easily read from the drawing. The value of a drawing for workshop purposes greatly depends upon the clearness and accuracy of the figures or dimensions and the skilful way in which they have been arranged. Often an occasional duplication of a dimension on different views will save much time in the works.

In cases where original drawings are not likely to be much used, it is the practice of many engineers not to ink them in. This, of course, necessitates more careful finishing in pencil. Indeed, the beginner should not be encouraged to do any inking in work until he has become fairly proficient in the somewhat difficult art of making a good pencil drawing.

The particulars as to the **scale** to which the drawing is made *must* always be clearly shown upon that drawing, not in order to enable workmen to "scale it," as sufficient dimensions should always be given to entirely obviate this. If there is any probability of the drawing being sent abroad where a different system of measurement is used, or to where it will be exposed to variations of temperature, the scale should always be drawn upon the drawing. Sometimes the scale of a working drawing has to be reduced to make it suitable for attachment to a specification, or some such purpose; in such a case, **proportional compasses** may be advantageously employed, the best practice being to locate the centres of the circles and curves and to ink the latter in direct, and then to proceed with the straight lines, avoiding the use of pencils as much as possible.

SHADE LINES AND LINE SHADING.[3]

31. Shade Lines.—The appearance of *finished drawings* (which are usually made to a small scale) is improved, and the true form of parts made more intelligible in a single view, by the use of shade or dark lines, which give an appearance of relief to the various parts.

Shade lines indicate the intersection of two surfaces, one of which is in the shade and the other illuminated. In arranging the shade lines, the parallel rays of light are conventionally assumed to come from the left and from behind (over the left shoulder) towards the object, their plans and elevations making angles of 45° with the vertical and horizontal planes respectively, their real inclination to the ground being 35°·15 nearly.[4] Thus, applying these rules to the body shown in Fig. 16, we have the back and right-hand edges *ab* and *bc*, also *ef* and *fg* of the projecting piece of the plan as shade lines; whilst the rules applied to the elevation give us the bottom and right-hand edges, *hi* and *ic'*, as shade lines. But, it should be explained, the line *hi* would not be a shade line if the body was actually resting on a horizontal surface, as the two surfaces would be in contact, and the upper not projecting beyond the lower. For these reasons *jk* is not a shade line, but *g'k* is. These rules applied to a case where there is a recess or hole, as in Fig. 17, give us the front and left-hand edges, *bc* and *ab*, as shade lines, the upper surface being in the light or illuminated, and the front and left-hand sides of the hole in the shade.

In dealing with **curved surfaces** shade lines are never used to denote their contour or outlines. Thus, in Fig. 18 the only

[1] This may be prepared by rubbing down a little colour from the cake of crimson lake.
[2] This may also be made by rubbing down a cake of the colour required, but most draughtsmen have the use of bottles of specially prepared red and blue inks.
[3] Articles 31 and 31a may very well be passed over by the student until he has reached and read Chapter VI.
[4] The cosine of the angle being obviously the $\sqrt{2} \div \sqrt{3}$.

EXAMPLES OF SHADE LINES AND LINE SHADING.

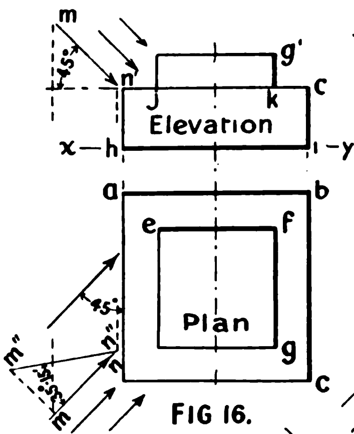


FIG 16.

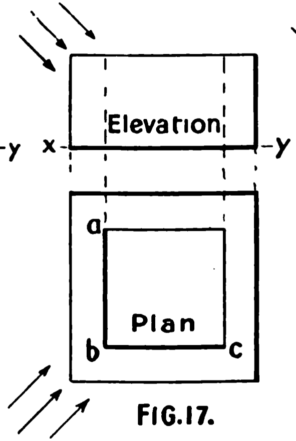


FIG.17.

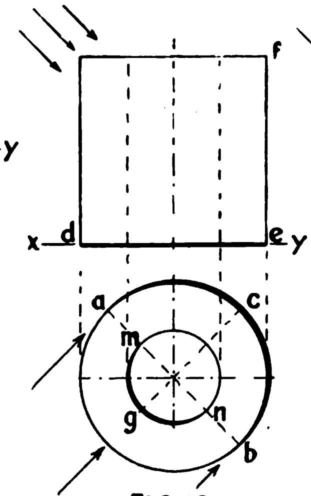


FIG 18.

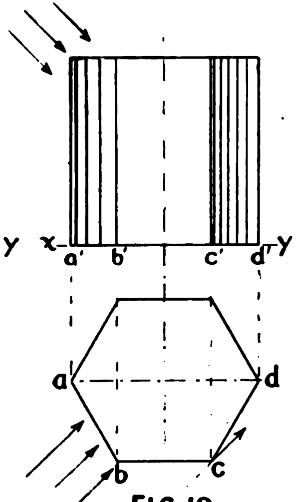


FIG.19.

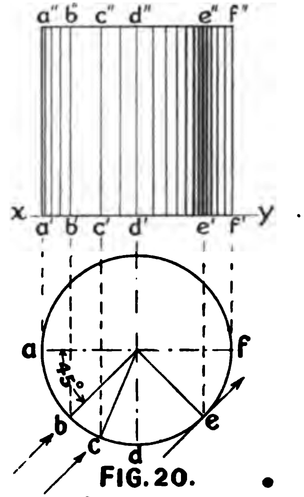


FIG.20.

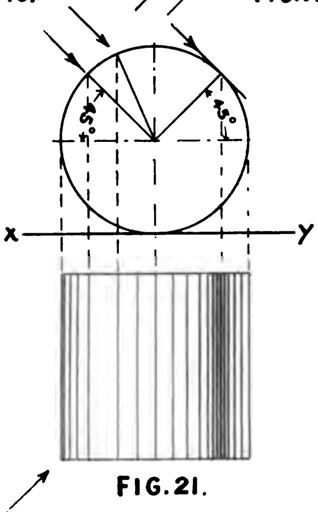


FIG.21.

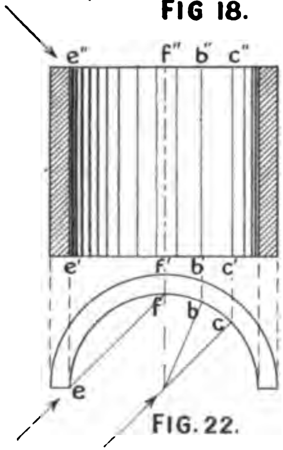


FIG.22.

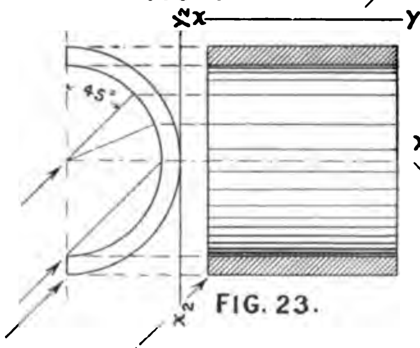


FIG. 23.

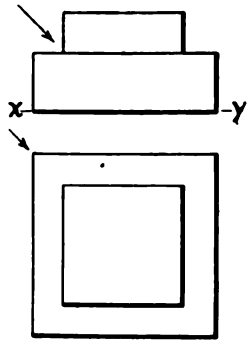


FIG. 24.

shade line on the elevation of the vertical cylinder is *de*, the line representing the solid's base, *fe*, being a boundary line of a *curved surface*, is not a shade line.

Now, the plan of the cylinder has a **curved outline**; and the rule relating to such cases is to make the **shade line** begin at the points *a* and *b*, at which the projections of the rays touch this outline, and let it gradually increase in thickness till its full strength is reached at *c*. Similarly, for the hole, the shade line increases in thickness from *m* and *n* to *g*.

The rules we have given relating to the rays of light we shall see are also concerned in the art of **Shading**, but, strangely enough, although generally followed by artists, many English engineers prefer to take the rays of light as shown in Fig. 24, where the rays *in plan* are *parallel* to those in *elevation;* this makes no difference to the elevation, but in plan the shade lines come in front, as shown, instead of at the back.

31a. Shading by Lines.—By shading a projection of an object its true form can often be rendered intelligible in a single view. For example, the shaded view of a cylinder explains itself. But, on account of the time and labour involved, **shading by tinting** is only rarely used, even in *finished machine drawings.* However, a similar effect can be easily produced by a few **shading lines,**[1] which are or should be drawn in accordance with the rules followed in shading proper. To commence with a simple example, that very often in many forms appears on machine drawings, we have in Fig. 19 a vertical hexagonal prism, with its front face in the light or illuminated. Such **surfaces parallel to the vertical plane** would **receive flat tints,** and **the nearer the surface is to the eye the lighter such tints** would be, and the shading lines would be equally spaced (between b' and c'), the spacing being increased on the lighter surfaces parallel to the plane of projection, and in the surfaces in the shade also receive flat tints, but the nearer such surfaces are to the eye the darker such tints are, or the closer the shade lines. Thus—

Surfaces in the light inclined to the plane of projection have given them graduated tints (represented by graduated lines, as shown between $a'b'$, Fig. 19), and as such surfaces recede from the eye the tints are made darker, or the lines closer together, as shown.

Surfaces in the shade inclined to the plane of projection also have given them graduated tints (or lines), and as such surfaces recede from the eye they are made lighter, or the lines further apart, as between c' and d', Fig. 19.

When two such **surfaces are unequally inclined**, the one upon which the rays impinge most directly is made lightest.

Curved Surfaces.—The above rules in the main are followed in shading curved surfaces. Thus in Fig. 20 we have the plan and elevation of a vertical cylinder upon which the light falls from $a'a''$ to $e'e''$, but most directly at the generator whose plan is *b*; this, therefore, as we have seen, should be the lightest part, but succeeding generators from $b'b''$ to $d'd''$ approach the eye, and according to what we have seen should therefore be increasingly lighter. So, in order to meet both these considerations, it is the practice to bisect *bd* in *c*, and make the surface between $b'b''$ and $c'c''$ the lightest; in fact, it is usually untinted, and remains white. Obviously, the darkest part of the cylinder is at $e'e''$, so that the shade and shading increase in depth from $c'c''$ to $e'e''$, and diminish from $e'e''$ to $f'f''$.

A horizontal cylinder, with its axis perpendicular to the vertical plane, is shown in Fig. 21, and the student will see that similar lines are used in arranging the shading. The case of a vertical hollow semi-cylinder is shown in Fig. 22, and, for reasons we have explained, the lightest part of the cylindrical surface is between the generators $b'b''$ and $c'c''$, and the darkest at the generator $e'e''$, the part between $e'e''$ and $f'f''$ being in the shade. Fig. 23 is a hollow horizontal semi-cylinder whose axis is parallel to the vertical plane, and the shading shown should now speak for itself.

[1] These are only used in connection with rounded surfaces on machine drawings.

32. Workshop Drawings.—The original drawings are kept in the drawing office for reference purposes, and copies only, produced in various ways, are used in the workshops. The most direct way of **copying a drawing** is to **trace** it on a sheet of tracing paper or tracing cloth, and, if more than one copy is required, the tracing is used to produce blue prints by sun-printing.

There are several **photo-copying processes** used for reproducing copies, or **blue prints**, by **heliography**, or **sun-printing** as it is called, in which the tracing is placed in front of, and in close contact with, a sensitized sheet of paper, both being clamped in a glass frame and exposed to the actinic[1] rays of light which, falling upon the tracing, pass through the transparent portions, decomposing the sensitized paper below, leaving the opaque lines upon the tracing undecomposed and transferred to the sensitized sheet. This sheet is then removed from the frame, and washed in water or certain solutions to remove the sensitizing matter and thereby develop the lines.

The various processes in use are the *Ferro-cyanide, Ferro-gallic, Platinotype, Zincographic,* and *Ferro-prussiate.*

The sun-printing process has the drawback of being somewhat slow, since it is mainly dependent upon the character of the natural light, and as this varies a great deal, so does the time taken to make the prints; but since the invention some years ago of the **electrical photo-copying apparatus**, in which electricity is used to produce the requisite light, engineers have had at their command a simple, handy apparatus which makes them independent of the weather, and in which prints may be made in two or three minutes. Perhaps the best-known apparatus of this kind is the one invented by Messrs. Shaw and Halden, and manufactured by Messrs. J. Halden of Manchester.

In **using the apparatus** the tracing and sensitized paper is laid upon a vertical semi-cylindrical glass plate, and a cover or jacket is then laid over the back of the sensitized sheet and firmly clamped by engaging with a rod. The cylinder is then turned into position, and the arc lamp lowered gradually down its interior, the speed of lowering being arranged to suit the exposure required of various sensitized papers.

32a. Tracing.—No small amount of skill is required to expeditiously make a good tracing. The beginner cannot do better than commence by drawing a number of straight lines and arcs of different thicknesses on tracing paper and cloth with his drawing pen and bow pen respectively. The ink may be introduced between the nibs of the pen by a pointed quill, or by a writing pen, care being taken to wipe the outside of the nibs, to prevent any ink from them touching the edge of the square, straight edge, or set square, for should the ink get in contact with these instruments it runs on to the paper and spoils the tracing. Care must be taken to preserve **uniformity of thickness** in the lines where required, and to make **arcs and curves flow into straight lines** without any apparent break,—in other words, to satisfy the geometrical condition for tangential contact. If the **ink does not freely run** on the tracing paper or cloth, a little powdered chalk may be rubbed over the sheet, or a drop or two of ox-gall may be added to the ink.

In commencing a tracing, be careful to pin the tracing paper over the drawing and on to the drawing board in such a way that the sheets are taut, and the principal line of the drawing is square with the working edge of the board. As a general rule it is best to draw all the lines that are in the direction of the length of the board first. In working down from top to bottom in doing this, many lines will probably be missed, but they will be picked up by working down a second, and even a third time if necessary. The transverse lines can then be drawn in the same way, and then any connecting arcs drawn, and the circles, if any, described.

[1] The action, as in photography, of the sun's rays in their chemical, as distinct from their illuminating and heating, effects.

Usually the first tracing made by a beginner is not much of a success, but by persevering in the way indicated he should soon become proficient. Great care, of course, has to be taken in writing the dimensions, to ensure absolute accuracy.

33. Tracing Exercises.—Beginners will find that the figures given for tracing in the B. of E. Examination papers at the end of the book, are the most suitable to commence on. After a little practice on these, he will be able to attempt the tracing of most of the drawing exercises given in the book, of course working on the simpler ones first.

CHAPTER III

SCALES, AND DRAWING TO SCALE

34. If we wish to draw the elevation of a machine whose height[1] is, say, 5′, and length 12′, upon a sheet of paper whose surface does not exceed two or three square feet in area, it is evident it would be impossible to make this drawing of the machine full size. Now, suppose we make a line 3″ in length on the drawing represent a foot on the machine, then a line 5″ × 3″ = 15″ long would represent the height of the machine, and one 12″ × 3″, or 36″ long, its length; and we should speak of the scale as being one of 3″ to the foot, and the *fraction of the scale,* as it is called (or representative fraction as it is sometimes called), would be—

$$\frac{3 \text{ inches}}{1 \text{ foot}} = \frac{3}{12} = \frac{1}{4}$$

In the same way: If $\frac{1}{4}$	inch	represented	1 foot	the scale	would be	$\frac{1}{48}$
" $\frac{1}{2}$	"	"	"	"		$\frac{1}{24}$
" 1	"	"	"	"		$\frac{1}{12}$
" $1\frac{1}{2}$	"	"	"	"		$\frac{1}{8}$
" 4	"	"	"	"		$\frac{1}{3}$
" $4\frac{1}{2}$	"	"	"	"		$\frac{3}{8}$
" 6	"	"	"	"		$\frac{1}{2}$

And if 1 inch represented 1 yard, the scale would be $\dfrac{1}{1 \times 12 \times 3} = \dfrac{1}{36}$

" 1 " 1 chain " " $\dfrac{1}{12 \times 66} = \dfrac{1}{792}$

" 1 millimetre represented 1 centimetre, scale would be $\frac{1}{10}$

" 1 " " 1 decimetre " " $\frac{1}{100}$

" 1 " " 1 metre " " $\frac{1}{1000}$

[1] The **dimensions** of machines, details, etc., are usually written in feet and inches. The former being indicated by the suffix ′, and the latter by the suffix ″. Thus, 5′ reads 5 feet, and 5′ $3\frac{5}{8}$″ reads 5 feet $3\frac{5}{8}$ inches. Further, 0·783″ reads decimal (or point) seven eight three of an inch, equal to $\frac{783}{1000}$ of an inch. When **metric measurements** are used the following abbreviations, *m.*, *dm.*, *cm.*, *mm.* respectively represent *metres*, *decimetres*, *centimetres*, and *millimetres*.

Angles are measured in *degrees*, *minutes*, and *seconds*. Thus 45° reads 45 degrees, and 20°, 40′, 50″ reads twenty degrees, forty minutes, fifty seconds.

Of course, whenever practicable, the drawing is made the same size as the thing to be drawn; the drawing is then spoken of as being *full size*. If the size of the object will not admit of its being drawn full size, then as large a scale as is practicable should be selected. This applies more particularly to detail drawings, where every minute feature must be clearly shown. The great size of some work necessitates its being set out in detail on large specially prepared boards, whilst, on the other hand, the details of watches, clocks, and small instruments can only be satisfactorily shown when drawn larger than their true size. In every case, whatever scale is decided upon, care must be taken to draw all parts of the object to the same scale, and thus get an exact, although a reduced or enlarged, representation of it. Scales should always be constructed and drawn at the foot of important drawings that are not fully dimensioned, so that the various parts may, with the aid of a pair of dividers, be scaled off, and so that any alteration in size, due to the shrinking of the paper, will affect both scale and drawing alike. These scales must be constructed and divided with great care and accuracy, and should be tested by measuring the same lengths from different parts of the scale. In drawing them, a very sharp pencil should be used, and when inked in the lines should be very fine.

35. Engineer's Scales.—Although most of the drawings made by the beginner will be **full-size** or **half-size**, for which any ordinary rule can be used, yet after some practice he will be called upon to make them to a smaller scale, such as $\frac{1}{4}$ or $\frac{1}{8}$ full size, or even less, so that he will require an instrument with these scales marked on it. Such instruments are called **Scales**, or **Drawing Scales**, and they can be had made of various materials, such as cardboard, vulcanite, boxwood, ivory, and steel. The ordinary lengths are 6″ and 12″, and they are made thin, and some are divided to the edge to enable a distance to be marked off from it with pencil or pricker; but a more accurate method is to take the distance off with dividers, as shown in Fig. 15, care being taken to lay the sides of the points on the scale or rule so as not to damage the points. (Refer to Art. 22.)

Vulcanite scales **should be avoided, as they expand and contract greatly with changes of temperature. On the whole, the best materials for them are *boxwood* and *ivory*. They are arranged with eight single reading open divided scales, two on each edge. The scales are 3″, $1\frac{1}{2}$″, 1″, $\frac{3}{4}$″, $\frac{1}{2}$″, $\frac{3}{8}$″, $\frac{1}{4}$″, and $\frac{1}{8}$″ to the foot, or $\frac{1}{4}$, $\frac{1}{8}$, $\frac{1}{12}$, $\frac{1}{16}$, $\frac{1}{24}$, $\frac{1}{32}$, $\frac{1}{48}$, and $\frac{1}{96}$ full size respectively.**

CHAPTER IV

HOW TO DRAW STRAIGHT LINES AND SIMPLE FIGURES

36. It is a waste of valuable time for the beginner to attempt to draw views, even of the simplest machine details, without some previous practice in drawing in a workmanlike way lines and circles (which are the component parts of such figures), and a few representative symmetrical figures. So the student is advised to carefully practise drawing the following progressive exercises, and after a few hours' practice he should be able to draw simple figures neatly and with accuracy.

37. Example.—**Straight Lines drawn with the Assistance of the T-Square.**—The student should patiently practise with his pencil and T-square in the following way:—

Commence by pinning the paper flat on the drawing board; this can best be done by first pinning one corner until the under-side of the pin-head is in close contact with the paper. Then place the back of the right hand upon the paper near this pin, and draw it diagonally across the sheet to the right-hand bottom corner, drawing the paper taut by the friction exerted. Hold this corner down by the thumb and fingers of the left hand, and insert a drawing pin in it as before described. The back of the right hand may then be placed at about the centre of the sheet, which is drawn diagonally to the right-hand top corner and pinned. Do the same with the remaining corner and the sheet will be as flat as it is possible to have it without damp stretching. The T-square can now be placed in position and held firmly by the left hand in such a way as to keep the stock in contact with the edge of the board, and the blade tight on the paper, as shown in Fig. 25. The pencil should be held between the first two fingers and thumb of the right hand, and kept in contact with the edge of the T-square, resting the third and fourth fingers on the square as the stroke is made.

The student must now aim at producing lines equal in thickness throughout their length, and, as the thickness and quality of a line depend upon the sharpness of the pencil and amount of unvarying pressure exerted upon it, he will understand that only practice will enable him to draw them with certainty and facility. Each line should be drawn the full length of the T-square, and several of each kind should be drawn; in fact, they should be drawn again and again till they can be freely produced at least equal in quality to those shown in the following figure (26), where it will be seen that A is a very fine line, suitable for centre and construction lines. This should be drawn with a very sharp chisel-pointed pencil, and should be so fine that a light touch of the indiarubber will clean it out. At B is a line sensibly thicker than the previous one, and suitable for the finished lines of a very small drawing. C is thicker, and suitable for ordinary drawing purposes. D is more suitable for working drawing of single objects, drawn to a large scale, and E is a suitable line for shade lines on drawings; this line is best drawn with three strokes of the pencil, as the pressure necessary with a point thick enough to produce it with one stroke would in most cases break the lead. When lines thicker than E are to be drawn, a good finish can only be given them by three strokes of the pencil; the two outside ones should be sharp and distinct, and the

distance between them decided by the thickness of the required line. In making the third stroke, the pencil should be turned sideways, so as to fill the space between the outer lines.

38. Defects in Lines.—The main defects in lines which should be avoided are: *Varying thickness,* caused by varying the amount

FIG. 25.—Showing how the T-square and pencil should be held.

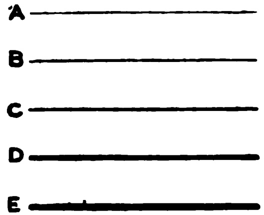


FIG. 26.—Thickness of lines.

of pressure exerted upon the pencil. *Want of sharpness,* the sides of the lines having a blurred appearance, caused by softness of lead or want of sharpness in the pencil. *Uneven colour,* due to unequal quality of the lead or paper, or uneven pressure upon the pencil.

39. EXAMPLE.—**Straight Lines, drawn with the Assistance of a Set-Square.**—The student should remember the instructions given for the previous example, and should now practise drawing similar lines with the assistance of one of his set-squares. The larger one had better be used, and the lines drawn its full length, at first to the right-hand side of the square as shown in Fig. 27 (and at A, Fig. 28) and afterwards to the left as shown in Fig. 29 (and at B, Fig. 28) in the direction indicated by the arrows. It will be seen that the left hand in each case is firmly holding the set-square and T-square together and on to the board in such a way that the stock of the T-square is kept closely in contact with the edge of the board. The remarks upon the previous exercise respecting the quality of the lines apply equally to this one, and the necessity of practising the drawing of these lines from both sides of the set square will be understood by the student after his first attempts, as he will find that to steadily move his hand about with ease, in the required ways, needs considerable practice.

40. Dotted Lines.—Dotted Lines are used on drawings either to indicate the line upon which a section has been taken or to mark the position of any existing part which is unseen; for the former, *dot-and-dash* lines, as at A (Fig. 30), are used, whilst for the latter **chain-dotted lines**, B, should be used. In the former case, A, they look best when the dots are equally spaced, and the short lines or dashes are equal in length, and about four or five times the lengths of the spaces; and in the latter case, B, when of equal length and equally spaced, the lines being made three or four times the length of the spaces, as shown. Obviously, if the dots are made shorter, they take a longer time to draw. The thickness of the lines, and the lengths of the spaces and dots, should be regulated by

the size of the drawing. A glance at some of the following lines, A to E (Fig. 31), will give the student some idea of what is considered good proportion, showing how they should vary in form with the thickness; and the student should patiently practise drawing

FIG. 27.—Using set-square with downward stroke of pencil.

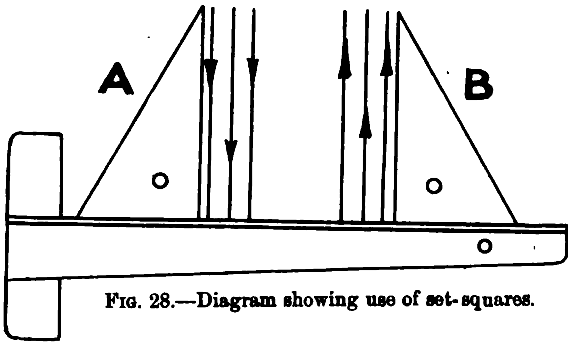


FIG. 28.—Diagram showing use of set-squares.

FIG. 29.—Using set-square with upward stroke of pencil.

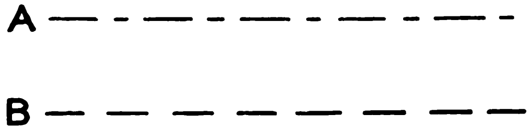


FIG. 30.—Dotted lines: different forms.

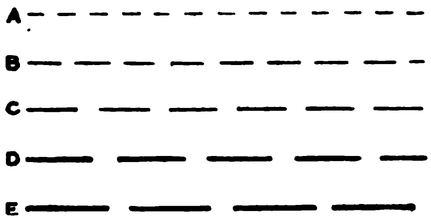


FIG. 31.—Examples of dotted lines: different thicknesses.

such lines until he can space them with a fair amount of neatness and facility.

41. Rectangles.—The student should now be in a position to draw some simple figures. Having practised on lines drawn in

the direction of the T-square, and at right angles to it, figures whose sides are made up of such lines should be easily drawn. So, by carefully working the following progressive exercises, which are very fully described, the student should make an important step in the practice of mechanical drawing.

42. EXAMPLE.—**To Draw a Rectangle whose Length (2″) and Breadth (1½″) are given.**—Draw, with the aid of the T-square, a very fine indefinite line AB, about 2½″ long, Fig. 32. With the aid of a rule and a pair of dividers prick off (Art. 22) the length CD equal to 2″, and between these two points draw a good finished line as shown. Then, with the aid of a set-square, draw from C and D very fine distinct lines perpendicular to CD and a little longer than the given breadth (1½″).[1] Now, prick off as before the point E (Fig. 33) from C, making CE equal to 1½″, the given breadth, and with the aid of the T-square, draw the finished line EF parallel to CD.

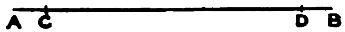


FIG. 32.—Construction of a rectangle. First step.

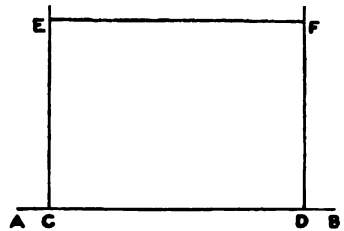


FIG. 33.—Construction of a rectangle. Second step.

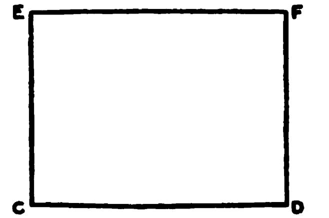


FIG. 34.—Construction of a rectangle. The complete figure.

The rectangle is completed by re-drawing CE and DF (Fig. 34), with the aid of the set-square, perpendicular to CD, being careful to regulate the thickness of the lines, so that they are the same throughout the figure, and removing with indiarubber the ends of the construction lines AC and DB, and those above E and F, leaving the rectangle completed as shown, care being taken not to remove the sharp corners formed by the intersection of the lines.

NOTE.—The student should always aim at constructing a figure by drawing the least number of lines possible; in other words, *a line should not be gone over twice if once will suffice.* As an illustration of this advice, with reference to the rectangle just drawn, many students would first have drawn the complete figure in fine lines, and then pencilled over each line to make it of the required thickness. Such a practice usually produces a poor result, as it is difficult to exactly cover the previous lines, and, further, it takes a longer time.

43. **Exercises upon the Use of Centre Lines.**[2]—*First Case.* **Figure Symmetrical about a Single Centre Line.**—Whenever a figure has more than one line each side of its centre, and is symmetrical about that centre, it is best drawn by commencing with the centre line. To illustrate this, let us proceed to draw the figure shown in the dimensioned sketch (Fig. 35).

Commence by drawing a very fine line AB (Fig. 36), with the aid of the T-square; then with dividers prickoff upon it two points C and D, 2″ apart. Through these points, with the aid of a set-square, draw two fine indefinite lines EG and FH. Then, with the dividers, prick off on one of these lines, say from C, the points J and K (Fig. 37), the opening of the dividers being ⅝″, equal to a half of the breadth (1¼″) of the given figure, and with the aid of the T-square draw through these points the finished full

[1] The student, after a little practice, will be able to estimate these distances and lengths to within a quarter of an inch, so that such lines need not be drawn much longer than their required length, to minimize rubbing out, but in no case should they be drawn too short at first, as any attempt at joining a length on is usually noticeable, and should be avoided.

[2] Centre lines should be very fine continuous ones, undotted, as at A, Fig. 26; then any part of them can be used to measure to or from.

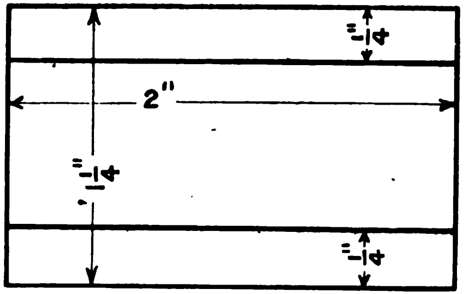


Fig. 35.—Rectangular figure, symmetrical about a centre line. The complete figure.

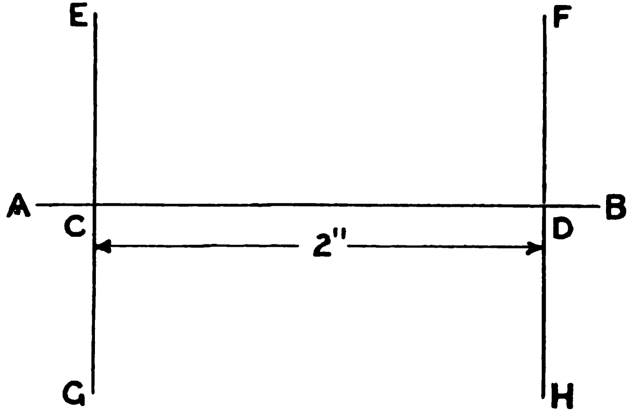


Fig. 36.—Rectangular figure. First step.

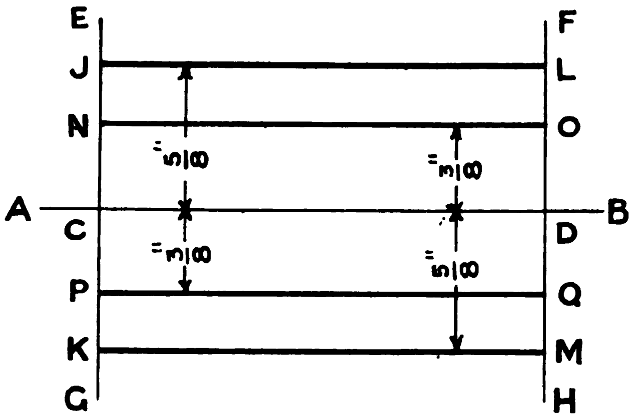


Fig. 37 —Rectangular figure. Second step.

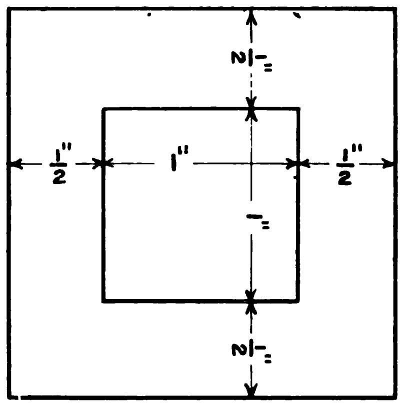


Fig. 38.—Square figure. Use of two centre lines.

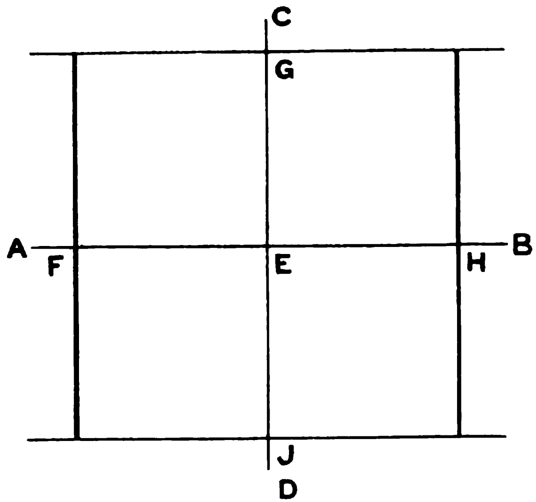


Fig. 39.—Square figure. Construction lines.

lines KM and JL. In a similar way mark off N and P from C, with the dividers open to $\frac{3}{8}''$, and through these points draw, in a similar way, the lines NO and PQ.

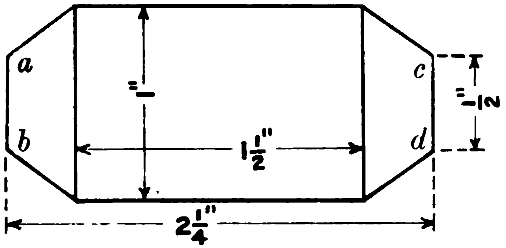


FIG. 40.—Figure symmetrical about two centre lines.

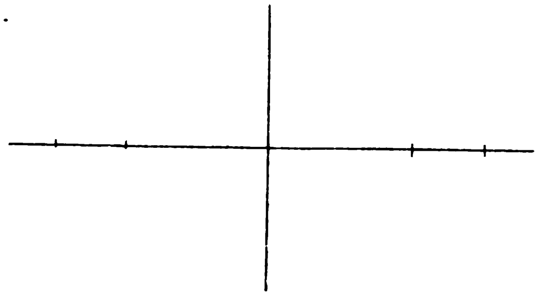


FIG. 41.—First step

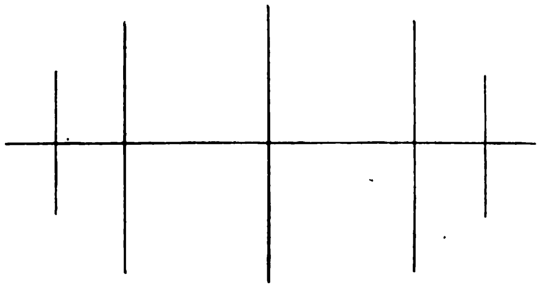


FIG. 42.—Second step.

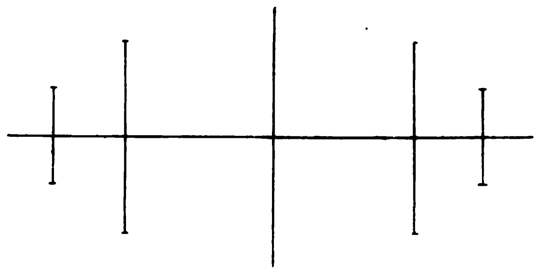


FIG. 43.—Third step.

The figure should now be completed by going over the lines KJ and LM with the pencil, taking care to give the lines the same thickness and finish as the others, and the figure will be now complete as in Fig. 35.

The projecting parts of the construction lines should now be rubbed out, as in the previous exercise, with indiarubber, the centre line AB being left projecting about a $\frac{1}{4}''$ beyond the figure upon each side.

NOTE.—The appearance and finish of the figure depends upon the lines being perfectly uniform in thickness and colour, and the student should constantly bear in mind the instructions previously given respecting the production of such lines.

44. *Second Case.* **Figure Symmetrical about Two Centre Lines.**—The figure No. 38 consists of two concentric squares which are symmetrical about two centre lines, at right angles to each other. So, first draw any two indefinite centre lines AB and CD, perpendicular to one another (Fig. 39), and intersecting at E; then, with rule and dividers, prick off from E, along the centre lines EF, EG, EH, and EJ, distances equal to half the side of the outer square, viz. 1", and complete the square as in the previous case. The inner square should be drawn in the same way, the construction lines removed, and the required figure completed as shown in Fig. 38.

45. EXAMPLE.—**Another Case of a Figure Symmetrical about Two Centre Lines.**—The figure to be drawn in this exercise consists of a rectangle, with a trapezoid at each end (Fig. 40). It will not be necessary to explain every step in the construction of the figure, as the student should by this time be familiar with the method of working from centre lines, and might now attempt to draw the figure in what appears to him the best way, with a hint that the small ends *ab* and *cd* of the trapezoid should be drawn before the sloping sides.

The figures 41, 42, and 43 show the steps in the construction.

These should speak for themselves now. Of course Fig. 40 shows the finished figure. But the student should not trouble about writing dimensions on his drawings yet.

CHAPTER V

CIRCLES, ARCS, AND LINES

46. As ordinary mechanical drawings mainly consist of combinations of circles, arcs, and lines, the art of correctly and neatly drawing a few of them in various positions in relation one to the others should be cultivated by the beginner; for if such lines are faulty in form and finish, or do not satisfy the geometrical conditions of proper contact, they spoil the appearance and detract from the value of any drawing upon which they appear.

As the student will have a great deal of work to do with the compasses, he cannot do better than carefully read the remarks upon their manipulation, etc. (Art. 17), before attempting this chapter. A few of the more important definitions and problems relating to circles and arcs are given here to help beginners, but for more complete information on these matters refer to the author's "Geometrical Drawing," pp. 61, etc.

47. *Definitions.*—**The radius of a circle** is a straight line drawn from the centre to its circumference.

A diameter of a circle is a straight line passing through its centre, and terminated on both sides by the circumference.

An arc of a circle is any part of the circumference.

A chord is a straight line joining the extremities of an arc.

A segment is any part of a circle, bounded by an arc and its chord.

A semicircle is half a circle, or a segment cut off by a diameter.

A sector is any part of a circle bounded by an arc and two radii drawn to its extremities.

A quadrant, or quarter of a circle, is a sector having a quarter of a circumference for its arc, and the two radii perpendicular to each other.

A sextant, or sixth of a circle, is a sector having a sixth of the circumference for its arc, and the two radii making an angle of 60° with each other.

An octant, or eighth of a circle, is a sector having an eighth of the circumference for its arc, and the two radii making an angle of 45° with each other.

A tangent is any line perpendicular to a radius at its extremity in the circle. A tangent touches the circle in a point, as at P, Fig. 44 (which is called the point of contact), where the line AB touches the circle, and it is perpendicular to the radius OP.

Point of Contact.—When two circles touch one another, they do so in a point only, called the *point of contact*, and the straight line which joins their centres passes through this point. Thus, Fig. 45 shows two circles, A and B, touching one another in the point P, which is the point of contact. It is only when this condition is satisfied that a part of one circle can be made to flow into a part of the other; the thick line in the figure shows how this condition must be satisfied.

To enable the student to correctly treat cases where circles are in contact with one another, and with straight lines, he should carefully study the following problems before attempting the exercises at the end of the chapter.

48. To describe a Circular Arc through Three given Points.—Let ABC (Fig. 46) be the given points. Join AB and BC, and bisect the lines AB and BC in G and D, and through these points draw perpendiculars intersecting in F. Then, with F as centre and radius FB, describe the required arc ABC.

49. To draw a Tangent to a Circle through a fixed Point in its Circumference.—Let B (Fig. 47) be the fixed point in the circle. Join B to the centre A, and through B draw CD perpendicular to AB. Then CD is the tangent required.

49a. To draw a Tangent to a Circle through a fixed Point without it.—Let the circle in Fig. 47 be the given one, and P the point.

Join the centre A with P, the fixed point without the circle, and bisect AP in E. With E as centre, radius EA, describe the semicircle AFP, cutting the given circle in F. Join PF. Then PF is the required tangent.

It is evident that in a similar way a tangent the other side of PA could have been drawn.

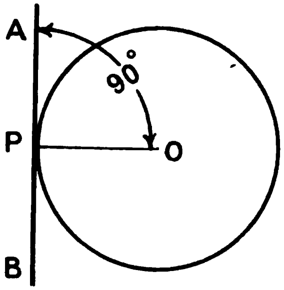


FIG. 44.—Circle and tangent.

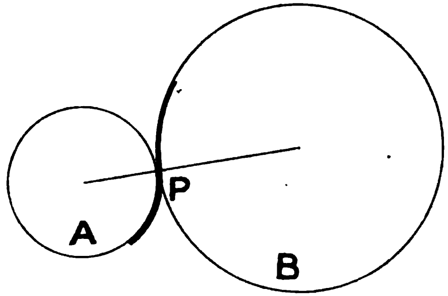


FIG. 45.—Point of contact of two circles.

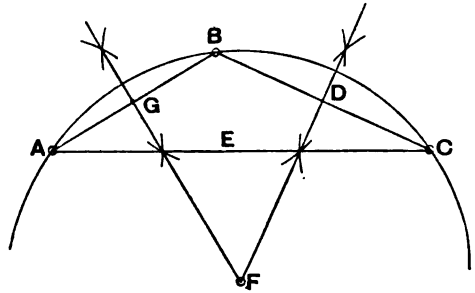


FIG. 46.—An arc described through three points.

NOTES.—1. The student will notice that if F be joined with A, the angle PFA will be a right angle, being an angle in a semicircle (Euc. III. 31). And FA will be a normal to the tangent at F.

2. The Euclidean geometry does not allow a tangent from a fixed point to a given circle to be drawn without first finding the point of contact as above, and the

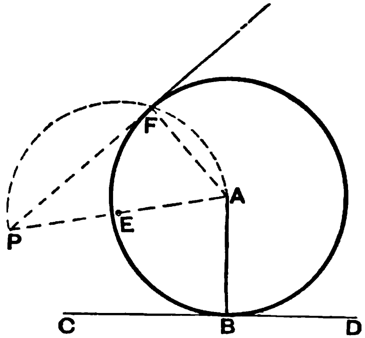


FIG. 47.—Tangents to a circle.

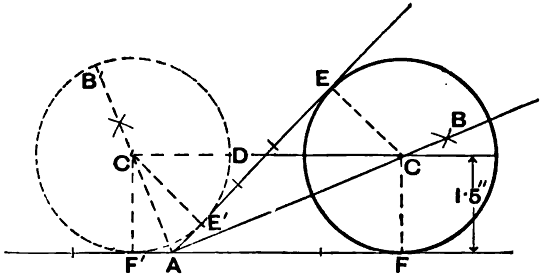


FIG. 48.—Circle touching two given lines.

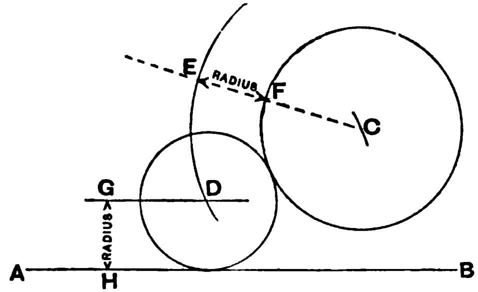


FIG. 49.—Circle of given size touching a fixed line and circle.

same remarks apply to the case of a common tangent to two circles, but for practical drawing purposes a tangent may be drawn from an external point to a circle, or a common tangent to two circles directly by carefully adjusting the straight-edge; and should the actual point of contact be required, a perpendicular to the tangent from the centre fixes it.

49b. To inscribe in a given Angle a Circle of given Radius (say 1·5").—Let EAF (Fig. 48) be the given angle. Bisect the angle

by the line AB, and draw CD parallel to AF and 1·5″ from it, intersecting AB in C. With C as centre, radius 1·5″, draw the circle touching the sides of the angle in E and F. The exact points of contact can be found by drawing from C the lines CE and CF perpendicular to AE and AF respectively.[1]

The dotted lines refer to a case when the angle is obtuse, and the same letters apply.

NOTES.—1. This is a problem often met with in mechanical drawing, when two lines are to be connected by an arc of a circle of given radius.

2. For most practical purposes a common tangent to two given circles (such as E′E, Fig. 48) can be drawn with a sufficient degree of accuracy by offering the edge of a square to the two circles and drawing a line to touch them, the points of contact being found by drawing perpendiculars from the centres to the tangent.

50. To describe a Circle of given Radius to touch a given Line and a given Circle.—From C (Fig. 49), the centre of the given circle, draw any line CE, cutting the circle in F; from F mark off FE, equal to the given radius, and with centre C describe the arc ED. At any point H in AB draw GH equal to the given radius EF, and perpendicular to AB. Through G draw GD parallel to AB, and cutting the arc DE in D. Then, with D as centre, radius EF, describe the required circle.

NOTE.—D is equidistant from line and circle.

51. To draw a Circle to touch Three given Straight Lines.—Let the given lines be AB, AC, and CD (Fig. 50), intersecting in A and C. Bisect the angle BAC by the line AE. The centre of the required circle must be somewhere in this line. Bisect the angle ACD by the line CF; the centre must also be somewhere in CF. Therefore it is in G, the intersection of AE and CF. From G draw GH perpendicular to AB and cutting it in H. With centre G, radius GH, describe the required circle or arc.[2] Then perpendiculars from G, such as GH, give the points of contact.

NOTE.—This problem sometimes occurs when a small bevel wheel is drawn.

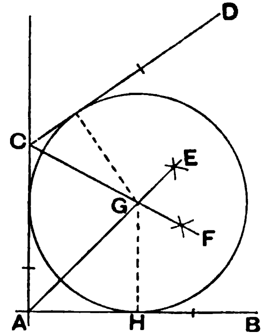


FIG. 50.—Circle touching three straight lines.

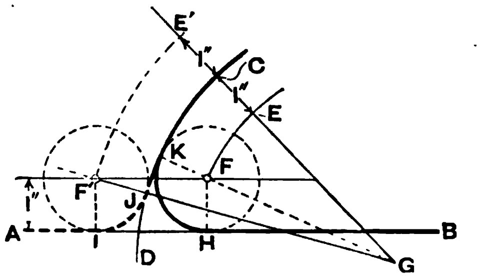


FIG. 51.—Arc touching straight line and arc.

52. To describe an Arc of a Circle of given Radius (say 1″) to touch a given Arc and a given Straight Line.—Let AB (Fig. 51) be the given line, and G the centre of the given arc. Draw GC, any line passing through the centre of the circle G, and cutting the arc in C. Mark off CE equal to the given radius of 1″, and with centre G, radius GE, describe the arc EF, and draw F′F parallel to AB and 1″ from it, intersecting EF in F, which is the centre of the required arc. With F as centre, radius FH, a perpendicular to AB, describe the required arc. Draw through G and F the line GK, cutting the circle in K. Then the points of contact are H and K.

If the arc were to touch the given circle externally, F′ would be its centre, and I and J its points of contact. The working is similar, and can be easily followed on the figure.

NOTE.—This problem occurs when a wheel with arms is drawn. HB is then the side of an arm, and the arc CK a part of the rim.

[1] AE and AF are two tangents to the circle from A, and they are equal to one another (Euc. III. 17).

[2] Three other circles can be drawn to touch the given lines, and one of them will obviously be contained by the triangle made by producing DC and BA till they meet.

53. Having studied the preceding problems, the student should be able to work the following exercises without further help. They should be carefully constructed from the dimensions shown, and not merely copied. Having pinned down a sheet of paper, the T and set squares should be carefully dusted, and the pencils and lead of the pencil bows to be used sharpened, and the latter adjusted so that the pencil and steel point are of equal length; the exercises can then be proceeded with.

EXERCISES.

1. Assume any point P in the given circle (Fig. 52), and draw a tangent at the point.
2. Through the fixed point P (Fig. 53) draw a tangent to the given circle.
3. In the angles ABD and CBD (Fig. 54) inscribe arcs of 1¼″ radius.
4. Describe a circle of 1¾″ diameter to touch both the line AB (Fig. 55) and the given circle.
5. Describe a circle touching the three given lines, BA, AC, and CD (Fig. 56), and mark the points of contact.
6. Describe a 2¼″ circle to touch both the given circles (Fig. 56A).

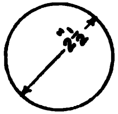


Fig. 52.

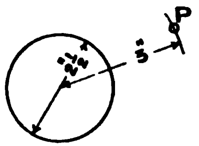


Fig. 53.

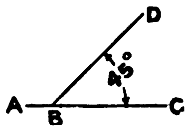


Fig. 54.

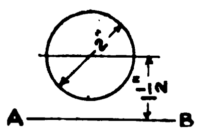


Fig 55.

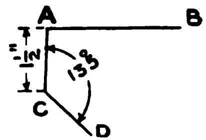


Fig. 56.

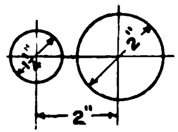


Fig. 56A.

CHAPTER VI

HOW TO COMMENCE A WORKING DRAWING

54. We will assume that the student has carefully read the preceding pages, particularly those from Art. 34 onwards, and that he is about to attempt a drawing of some simple object. Now, before he can do this intelligently, it is obvious that he should have a fair acquaintance with elementary projection, such as is taught him in his geometry class, so, if on joining a class in Machine Construction and Drawing he has not had some training in solid geometry, he will doubtless be recommended by his instructor to take up the study of that subject concurrently with his course in drawing.

In most schools or Institutes this can be easily done, as the geometry and machine drawing classes are often so arranged that they can be attended the same evening or day. However, for the benefit of those who may not be able to attend such classes, or have not the help of a teacher, we will proceed to briefly explain how an object may be drawn in plan and elevation, for the shape and proportions of most simple solids can be completely shown by drawing two views only, namely—

55. Plan and Elevation, called their *projections*. The terms "plan" and "elevation," as applied to the representation of an object, are fairly well understood in a general way. Thus we speak of the elevation of a house, meaning the view we get by looking at its front, back, or sides. By such a view we see its height and breadth, and the height of everything shown is found on this *elevational* view. Again, we speak of the plan of a plot of ground. This view, of course, shows its length and breadth, and the distance it may be from some landmark. In the same way the plan of a house or any object is the view we get by looking down on it from above. All this and much more can better be made clear by referring to an example; and as first steps cannot be made too easy, the subject frequently presenting considerable difficulties to beginners, the tyro cannot do better than take a sheet of drawing paper and any rectangular solid, such as a box or a book, and work out the following simple exercise:—

Let BACD (Fig. 57) be the sheet of paper. Draw across it any line XY (this may be done in the ordinary way with the T-square), and place the bottom (EFGK) of your box on the paper, so that one of the long edges, EK, is resting on XY. Then bend the part of the paper BD about the line XY, as shown, until it touches the back of the box EKIJ. If, when the paper is in this position, a pencil point be drawn round the box, marking the lines EFGKIJ, we shall have on the horizontal plane (XYCA) a plan EFGK of the box, and on the vertical plane (XYDB) an elevation EKIJ. Now, let us suppose that we are to draw the plan and elevation of the box in its present position in the ordinary way. Begin by drawing XY (Fig. 58) with the aid of the T-square; then construct EKIJ (the elevation), a rectangle, making EK equal to the length of the box, and EJ equal to its thickness, remembering that EK must rest on the ground line (XY), as the box is resting on the ground (horizontal plane), and that as it is touching the vertical plane, the plan, which may now be projected (carried down) from the elevation, must be drawn showing the back EK of the box touching XY. Of course, all the lines on the plan and elevation are drawn with the assistance of the T-square and the set-square S. The student will notice that in this case the plan might have been drawn first, and the elevation projected from it. That is to say, this is a case where either the plan or elevation may be first drawn. (Cases will occur directly where this is not a matter of choice.) It will now be seen that in Fig. 58 we have represented the form and position of a body which possesses three dimensions (namely, length, breadth, and thickness) upon a plane having only two dimensions, namely, length and breadth. The student should now bend the paper (Fig. 58) about its XY so that the two parts are at right angles, as in Fig. 57, and then imagine that

the box is in its place, as it is shown in that figure, for beginners frequently fail to make much progress owing to their inability to exercise their imagination in this way.

As a further exercise we may draw the plan and elevation of a rectangular block in such positions as shown in Fig. 60, where it will be seen that the two views are separated by the distance *aa′*, and to enable the student to see what bearing this change of position has upon the previous case we will proceed to work a little problem which shall be a distinct step in advance of the previous study, but, nevertheless, one that ought to be readily understood. The problem may be stated thus:—

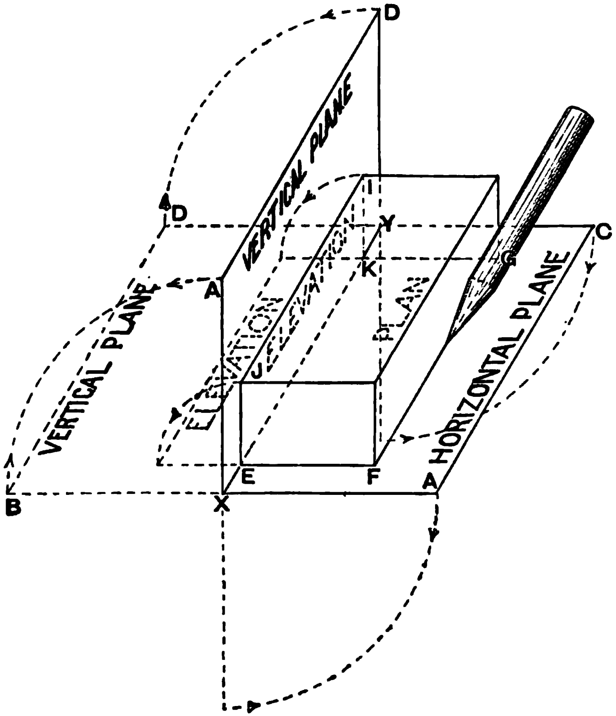


Fig. 57.—Relation of plan to elevation.

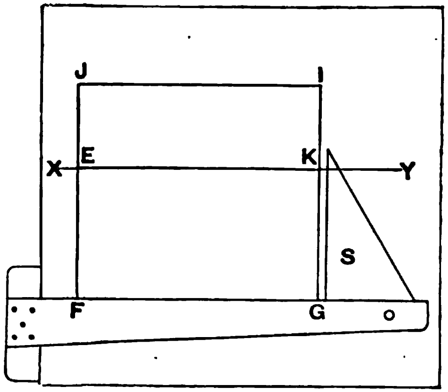


Fig. 58.—Projecting one view from the other.

56. To draw the Plan and Elevation of a Rectangular Block 9″ long, 6″ wide, and 3″ thick, when a 9″ × 6″ Face is horizontal, and 1″ above the H.P. (or Ground), and one of its Sides is parallel to the Vertical Plane, and 2″ from it. (Scale, half-size.)—First draw across the paper a line, and mark it XY[1] (Fig. 59). Then fold or bend the paper about this line, as in the previous study, and as shown in Fig. 59, and place the block on something 1″ thick; it will then be the right height above the ground, or horizontal plane. If we now move it till its back face is parallel to the vertical plane, and 2″ from it, the block will be in the required position. The figure clearly shows this position, and at this stage it will be instructive to compare this problem with the previous study (assuming that the box and the block are the same size). It will be noticed that the plan in Fig. 59 is the same shape as the plan in Fig. 58 (this must be so, as both solids are horizontal), but is 2″ distant from XY (that is, 2″ from the V.P.), and similarly with the elevations, they are the same shape. The one in Fig. 59, being 1″ above XY, shows that it is 1″ high. Of course it will be noticed that the lines (projectors) connecting the block with its elevation are perpendicular to the V.P., and also the lines connecting the block and the plan are perpendicular to the horizontal plane. The

[1] This is the ground-line, as it is called; it is invariably marked XY in geometry.

figure also shows by dotted lines the paper folded (constructed) back into its proper (normal) position, and the dotted elevation shown will be seen to be *in the same straight line with the plan, perpendicular to the ground-line* (XY). Thus, when the projections of an object are drawn, we always have the *plan and elevation in the same straight line perpendicular to the ground-line.*

To make this second study complete, let us suppose that we, knowing exactly how the views will appear in shape and position, wish to draw in the ordinary way the projections of the block to satisfy the problem. The first thing to do is to draw XY, the ground-line (Fig. 60). Then, as in this case we can first draw either projection, let us start on the plan. Remembering that the block is 2″ from the V.P., we draw a line *ab* parallel to XY and 2″ below it, and on this line we construct the plan, which of course is a rectangle, whose length is 9″ and breadth 6″. Then from each end of this plan draw a projector perpendicular to XY; between these projectors draw *a′b′*, the bottom of the elevation parallel to XY and 1″ above it, and on this line complete the rectangle, whose breadth is 3″ (the block's thickness), which forms the elevation. The projectors are best drawn undotted, but much thinner than the lines that form the projections.[1]

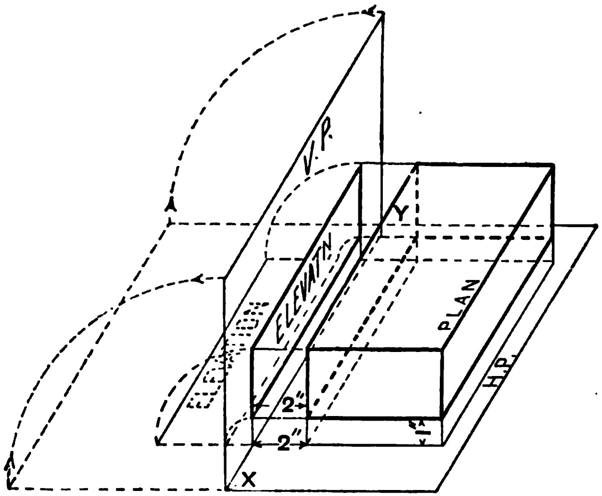


FIG. 59.—Block in position, between folded drawing paper.

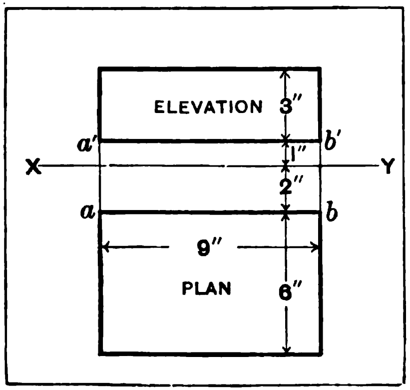


FIG. 60.—Projections of a rectangular block.

This completes the projections, and the student would do well to repeat the operation explained in the previous study, and try to imagine that the solid itself is standing over the plan, and in front of the elevation, as shown in the figure.

NOTE.—Before leaving this study, we might notice that the line *a′b′* on the elevation represents the bottom of the block, a horizontal surface, and a surface perpendicular to the vertical plane. The student will directly better understand that the projections of all surfaces perpendicular to a plane are straight lines on that plane. Thus the line *ab* on the horizontal plane is the plan of a vertical side.

57. End Elevations and Sections.—Let us suppose we are looking at the rectangular block (Fig. 61) in the direction of the arrow B, the view we then get is called an *end elevation*, and it may be shown as at E, where the figure is obviously constructed with the assistance of the plan, the 3″ height being marked off with the dividers. It is generally more convenient to place

[1] In an ordinary mechanical drawing the projectors are not allowed to remain; any that may have been drawn as a matter of necessity or convenience being rubbed out.

this view by the side of the elevation, as shown at F; the view is then projected from the elevation as shown, the 6″ breadth being marked off with the dividers or found by using the arcs *fm* and *hn*. If we were to cut through the solid with a vertical saw-cut along the line CD in plan, the true shape of the cut would be a vertical *section* (a section on the line CD as it is called) of the solid. This is shown at G in the position which is usually most convenient in relation to the elevation. It is drawn in the same way as the end elevation F.

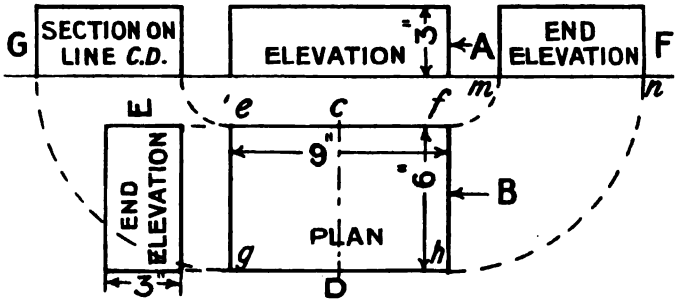


FIG. 61.—Projecting sections and end elevations.

We may now proceed to review the salient points as they would probably present themselves to the student who is about to commence a working drawing. He should begin by making up his mind as to how many views of the object he intends to show, bearing in mind that the drawings should clearly represent the object in such a way that its true dimensions and the form of every detail are shown. So long as this is satisfactorily accomplished, as few views as possible should be drawn. Two views at least are always required, and these may be an *Elevation* (which shows length and height) and a *Plan* (which shows length and breadth).

Or the front elevation and an end elevation may be used to obtain a similar result. But three views, namely, a *Front elevation*, an *End elevation*, and a *Plan*, are generally shown, with sufficient *sectional elevations* and *sectional plans* (part section and part elevation, and part section and part plan respectively) to make the external and internal form or construction of the object quite clear. The use of dotted lines, as in the end elevation at MM_2 KK_2 (Fig. 63), for indicating the position of unseen parts, should as a rule be avoided as far as possible; but a judicious use of a few of them may save the making of another view, provided always that they do not impair the clearness of the view upon which they are placed.

Dotted lines should not be used for unseen parts in highly finished coloured drawings, but only for working drawings. In cases where the object to be shown is symmetrical about the centre line, it is usual to show one half of the view in elevation, and the other half in section, as in the sectional elevation of the coupling (Fig. 98, Chapter IX.).

The section may extend slightly beyond the centre line, or may finish at it; in either case a black line is used to terminate the section. This saves the making of a separate sectional view.

Although it is obviously desirable to limit the number of views of an object, as previously explained, care must be taken not to carry this too far; as in the case of a complicated object, say a casting, much time is often spent by the pattern-maker and others in trying to read a drawing, where an additional view or section would have enabled the trained eye to see at a glance a mental picture of the required object.

It is usual to arrange elevations above plans, or sectional plans, when convenient; but in all cases the views must be arranged so that the relation between two adjoining ones may be readily recognized, and so as to facilitate their being properly projected one from another.

Having decided upon the number of views to be shown, it is usual to take a spare piece of paper, and to roughly sketch upon it the views decided upon in their relative positions one to another, and to mark upon each the overall sizes, as in Fig. 63.

The size of the sheet of drawing paper to be used should also be marked upon it, and allowances made usually from ¼" to ½" on each edge for cutting the drawing off square, and from 1" to 2" should be allowed inside the cutting off line upon each edge for a margin. If the paper has been damp-stretched, sufficient margin must be allowed to enable the drawing to be cut off clear inside the adhering edge. From 1½" to 3" is usually allowed between the border lines and the right- and left-hand views, and from 1" to 2" horizontally between separate views. These amounts must be added together, and subtracted from the length of the sheet; and then the dimensions of the longest line of horizontal lengths of the various views must be added together. By comparing this with the space remaining for them upon the sheet, the scale to which the views can be drawn may be decided upon.

After a scale has been assumed for the horizontal line of views, the longest line of vertical dimensions must be checked against this in a similar way to see if the scale is suitable.

All drawings forming one set should have equal outer margins, and as far as possible equal margins between the views.

Having arranged the positions of the views upon the sheet, and the scale to which they are to be drawn, the next thing to be done is to draw the cutting off and border lines upon the sheet, and then the centre lines of the various views. The positions of these can be readily ascertained from a rough sketch used to adjust the spacings, and they should be carefully marked out; and after this has been done, the various views may be commenced. Of course, these remarks are for the guidance of the young draughtsman. The beginner will always have plenty of paper to practise on, and need not trouble about the spacing out.

It is impossible to lay down any fixed rule as to what view should be first completed; in fact, it is usually the practice to work upon two or three views at the same time, drawing some part upon all views first, and then adding another part to these, and so on. But generally any known portion, such as the size of a shaft, stroke of a part, leading centres or outline is first drawn; and *always* the view from which the greatest number of parts of other views can be projected, or the greatest amount of information obtained (frequently a section) is then proceeded with; an axiom being to put in outside sizes of work definitely first, and to fill in all smaller details, as bolts, rivets, studs, nuts, keys, cotters, etc., afterwards. In the case where a part has a circular form, *the circles should be drawn first*, and the other views projected from them, and when a number of similar parts, as rivets, bolts, and nuts, occur, it is best to put in the small circles of the entire number first, with one setting of the compasses, and then the similar lines of each. This will take less time than if each one is completed singly, and ensures a more uniform result.

It is also usual to show upon working drawings, bolts, nuts, pins, rivets, studs, keys, cotters, rods, shafts, spindles, springs and levers in elevation, even when the section plane passes through their axes. The reason being that it is less trouble to show them in elevation than in section, and it renders the drawing more clear. But all these matters can now be more conveniently dealt with as we proceed to explain how drawings of a few simple objects may be made, starting with a very easy example and selecting others so that they may gradually present to the student further features and expedients in a progressive way.

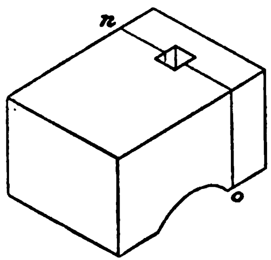


FIG. 62.—Isometric view of bench block.

58. Drawings of a Cast-iron Bench Block.—The sketch, Fig. 62, shows the form often given to a bench block or anvil, such as is often used in an engineer's fitting shop. Cast Iron is used for the block in preference to Wrought Iron, as it is much cheaper in first cost, and, being harder, is not so easily injured by a blow. The flat surfaces may be planed, but it is sometimes used rough as cast. In this and the following exercises, the views and scale selected are so arranged as to enable the object to be drawn upon a half-imperial sheet of paper, viz. 22" × 15".

As a **drawing example**, the four views of the block shown in Fig. 63, viz. a front elevation, a plan, an end elevation and a section on the line *no* taken transversely through the centre of the hole and looking to the right (the left-hand portion being removed), are to be drawn full size.

So commence by placing a sheet of paper on the drawing board and pin it down taut and flat, as explained in Art 37. This being a beginner's exercise, we need not trouble very much about spacing out the views of the block we wish to draw, as previously

explained. The student who has followed the previous exercises will by this time be fairly able to manipulate his instruments

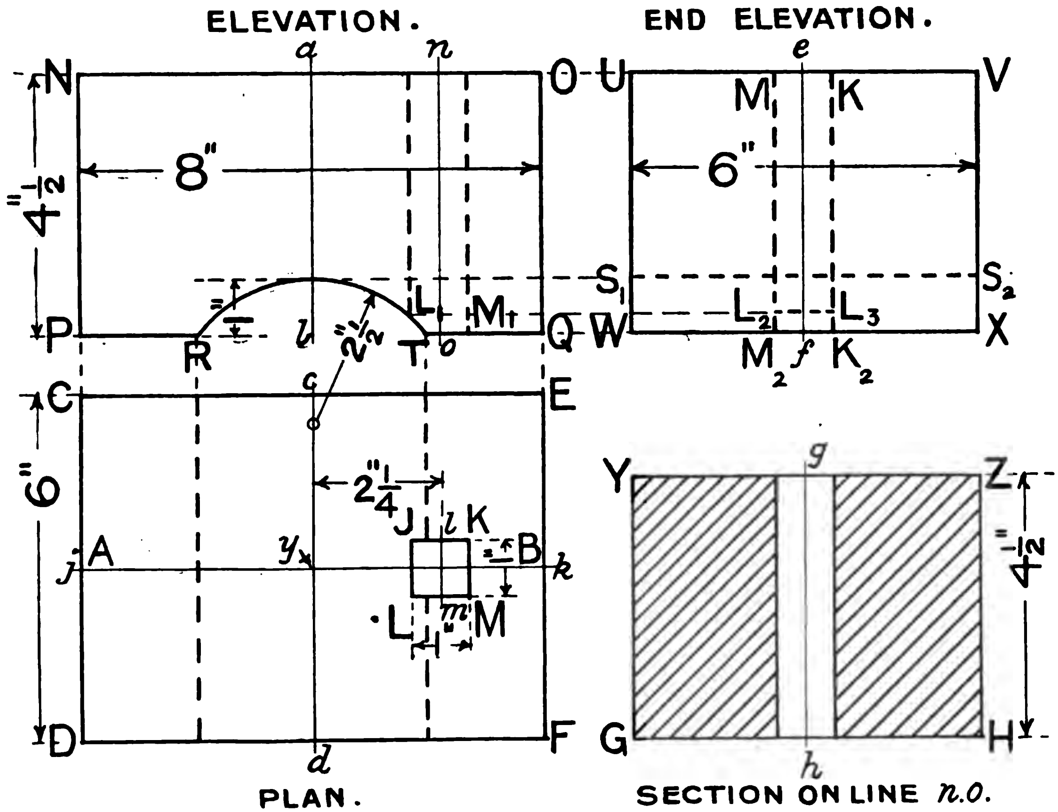


FIG. 63.—Four views of a cast-iron bench block.

correctly, and by the exercise of a little intelligence he will easily draw the plan and elevation of the block; so, bearing in mind the hints previously given as to which view to draw first, it will be seen that this is a case where the plan should be first set out. Then start by drawing the centre lines *jk* and *cd*, intersecting in *y* (Fig. 63), in suitable positions. The length of the block should be first set out by pricking off *yj* and *yk* with a 4″ opening of the dividers, the scale being full size. The T-square is then drawn down to about $3\frac{1}{4}''$ below *jk*, and the 60° set-square is placed upon it and brought into position so that the pencil will be in line *k*. The line is then lightly drawn downward, nearly [1] to the T-square; and the set-square is then slid along the T-square, and a line drawn through *j* in a similar manner. Next prick off with the dividers *c* and *d*, 3″ on each side of *y*. The T-square is then raised to the lower mark D, and the finished line DF is drawn carefully, once and for all, between the two vertical lines previously drawn. The T-square is then raised to the upper mark C, and a similar finished line CE drawn through it. Then rub out the extra portions of the lines at CD and EF and rule the vertical lines in, similar to the finished horizontal ones. Next draw the vertical centre line *lm* of the

[1] As we do not know exactly where to stop, we always rule it lightly and too long, and rub out what we do not require after its desired length has been obtained. This is much better than to rule a line too short, and to join a piece on to make it of the required length, as the joint always shows.

hole in the block, which will be $2\frac{1}{4}''$ from the centre of the block; and take the dividers and set them carefully to $\frac{1}{2}''$, and prick off points in the sides of the square from the intersection of the centre lines of the hole, and pencil in the sides JKLM of the square in the same way as the outline of the plan was done.

The elevation may now be proceeded with by first drawing an indefinite line PQ, a suitable distance from CE, and a similar line NO at the top, $4\frac{1}{2}''$ from it. The side lines PN and OQ may now be projected from the plan and drawn their finished thickness. The arched opening RT may now be drawn; first mark up centre line *ba*, the height (1″) of the arch above the bottom of the block, and set the pencil compasses to an opening of $2\frac{1}{2}''$ (the radius of the arch), and describe the arc RT as shown. Then project up the centre line of the hole from *lm*, drawing *no*, and making it about $\frac{1}{4}''$ longer top and bottom than the elevation. From J and K in plan project points on to the bottom of the block and line of arch, as L_1 and M_1; through these points draw vertical finished dotted lines as shown, from bottom to top of the elevation, to indicate the position of the hole.

To commence the **end elevation** project two indefinite lines UV and WX from the top and bottom of the elevation respectively, and draw the centre line *ef* in a suitable position. Then mark off 3″ each side of this line and draw the finished sides UW and VX, completing the outline as before. To indicate the position of the square hole on this view set off *e*M and *e*K, $\frac{1}{2}''$ each side of *e*, and draw the dotted lines MM_2 and KK_2. The dotted line S_1S_2 and L_2L_3 are projected from the elevation, and indicate the position of top of the arch part and the intersection of the arch with the side of the square hole respectively. The **section on line** *no* is drawn in a similar way about a centre line *gh*, the bottom GH being projected preferably from DF of the plan, and the sides YG and ZH from UW and VX respectively. Of course, the height GY is $4\frac{1}{2}''$, the same as that of the elevations. As we are looking at the section from the left, we shall see the right-hand side of the section.

The parts actually cut through by the section plane should be section-lined as shown, and as described in Art. 39. And the section lines on both right- and left-hand side of the hole should be drawn sloping in one direction only, as it is one piece of metal.

The section lines used to indicate cast iron are continuous ones (Fig. 69); as shown, they are drawn with the 45° set-square resting upon the T-square. The distance between them, or pitch of the lines, is a matter of taste, and should vary with the size of the part to be sectioned; in this case lines $\frac{1}{10}$th of an inch apart may be used. They can be drawn by judging the distances by the eye after a little practice, or a line can be drawn at right angles to the slope of the section lines, across the figure to be sectioned, and equal spaces set off upon it by ticking them off from a scale of equal parts, or by using a pair of dividers. To finish the drawing, carefully clean off any matter or lines not required, but the centre lines should be left, projecting about $\frac{1}{4}''$ beyond the boundary of the view they are shown upon. The **dimensions** need not at present be shown on the drawing. The title of the drawing should be neatly written (printed) by hand, at the top of the drawing, making it clear and brief.

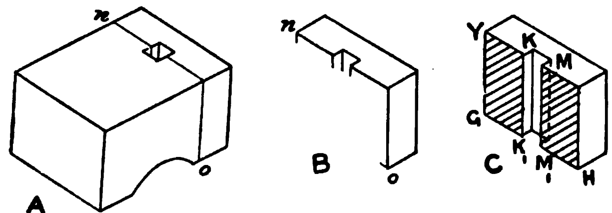


FIG. 64.

If the beginner has any difficulty in realizing what the section on line *no*, or any other section, shows, he is strongly recommended to make a kind of perspective sketch of the object, somewhat like that shown at "A" in Fig. 64, or better, if he will take the trouble to cut the object out in yellow soap, or mould it with putty or modelling clay. It need not be to scale, but should be roughly proportionate in size. This model he can cut in the desired position to enable him to realize what shape the section would be. If he uses a sketch, and has difficulty in deciding how the part cut by the section plane will appear, let him place the section line upon his sketch in the desired position, as *no*. Then rub out the forward portion (that to be removed) up to the section line, as

shown at "B," and then try to complete the sketch "B," obtaining the data necessary to enable him to do so from the other views of the object. For instance, knowing the block to be rectangular with parallel sides, he can add to "B" the lines YG and HG, Fig. "C." Then he knows from the elevations that the hole goes right through parallel to the sides, so he can draw the lines KK_1 and MM_1, indicating the cut hole. Of course this is only a sketch, but a student should have no difficulty in identifying it with the section on line *no*, as given in Fig. 63.

59. To draw a Section of a Wrought-Iron Beam or Joist.—Fig. 65 is a finished drawing of the section of the beam, drawn in a conventional way to a scale of one-half full size, and fully dimensioned. After studying the previous exercise, each step the student should take in making this simple drawing should be obvious; indeed, all that he should require is a hint or two to

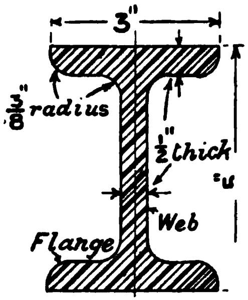


FIG. 65.—W. I. beam. Finished section.

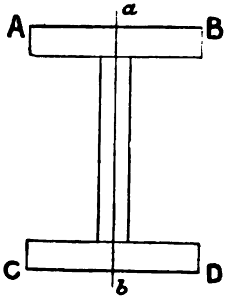


FIG. 66.—Section of W. I. beam. First step.

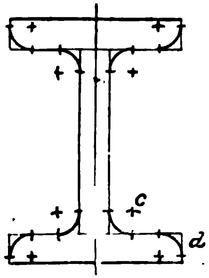


FIG. 67.—Section of W. I. beam. Second step.

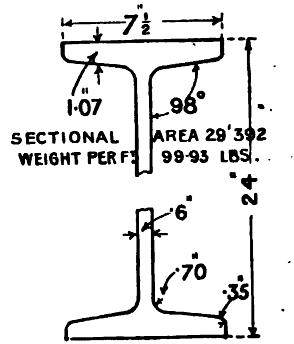


FIG. 68.—Section of maximum size standard beam.

enable him to go about it in a workmanlike way. The section being symmetrical about a centre line, this line should be drawn first, as *ab* (Fig. 66), and the rectangular outline of the section drawn as shown in the figure.

It will be noticed that the only lines in this figure that can be drawn in a finished state right off are AB and CD. The next step is to describe the arcs,[1] having previously found their centres as indicated at *c* and *d*, Fig. 67 (these centres can, with ordinary care and a little practice, be found by trial). The drawing then appears as shown in Fig. 67. All that now remains to be done is to carefully join the arcs and complete the outline with lines of uniform thickness throughout. The figure may now be *cross-hatched* or *section-lined*. The conventional lines in this case (as the material is wrought iron) are alternately thick and thin as shown in the Fig. 65. (Refer to Art. 61.) These sections are now *standardized*. (Refer to Art. 60.)

60. British Standard Beam Sections.—The form given to the beam section in Fig. 65 is conventional, it being a convenient one for drawing purposes. Formerly there was a great want of uniformity in the relative thickness of flanges and web, and also

[1] It will be noticed that the radius of the arcs is three-fourths the thickness of the metal. It should be explained that the actual radii vary with different makers, and in most cases the flanges are slightly taper in thickness (as shown in Fig. 68); but for drawing purposes the above proportions may be used, and the flanges made of uniform thickness.

of the radii of the fillets and edges, to say nothing of the amount of taper given to the flanges;[1] but in 1904 the Engineering Standards Committee published their Report on the Properties of British Standard Sections,[2] in which all the sections commonly used by ship and bridge builders, etc., are standardized. Fig. 68 gives the standard dimensions for the largest beam section, which is shown here as an example of a standardized section.[3]

61. Sectional Shading or Lining for Various Materials.—Fig. 69 shows the sectional shading that is very generally used to indicate the materials used in engineering work. They speak for themselves.

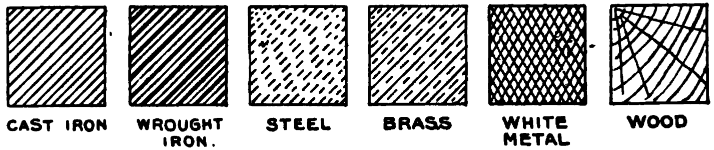


Fig. 69.—Conventional sectional lining for various materials.

62. Drawing of a Stuffing Box Gland. (Scale, full size.)—Figs. 70 and 71 show, in elevation and plan, a gun-metal stuffing box gland (fully dimensioned) for a $2\frac{1}{2}''$ piston rod or valve spindle.[4]

In commencing a drawing of these views, the student will first set out the centre lines *ab* and *cd*, as the object is symmetrical about these lines. Now, as matter of practice, as has been previously explained, whenever one of two views of a body or part of a body is circular in form, that view should be drawn first. So mark out centre lines for the holes A and B (Fig. 72), and describe the four circles in plan to the dimensions shown, giving the lines their finished thickness. Then, with 1″ radius, arcs may be drawn about the centres of the stud holes A and B with a light line, also arc DJ, of $2\frac{1}{16}''$ radius, about centre K, then tangents such as CD can be drawn, and the plan completed (as in Fig. 71) by going over the arcs EF and GH, etc., making them uniform in thickness with the other lines.

The elevation presents no difficulty, and should be easily drawn now.

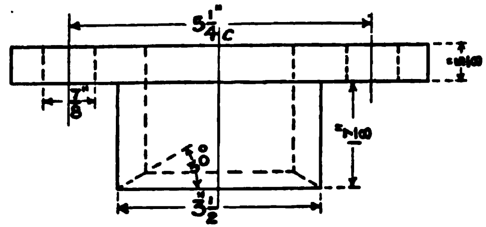


Fig. 70.

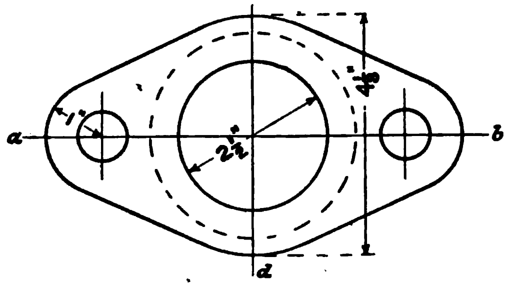


Fig. 71.

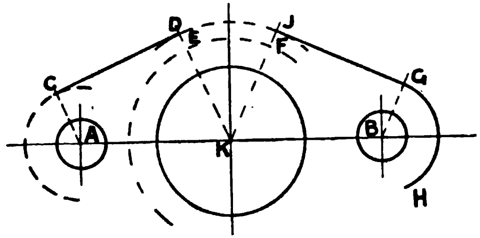


Fig. 72.

[1] Largely due to the many Continental sections on the market made to metric measurements.

[2] Published by Crosby, Lockwood & Son, price 3s. net.

[3] For further information relating to section of bars, etc., refer to Chapter XI.

[4] For particulars relating to stuffing boxes, etc., see Arts. from 63.

At this stage, a good exercise on the above would be *to draw a section of the gland made by a plane, cutting it in halves through the line ab* (Fig. 71).

Obviously, its outline would be similar to the elevation (Fig. 70).

EXERCISES.

Drawing Exercise.

1. Make a drawing of the beam section, Fig. 68. *Scale, quarter full size.*

Sketching Exercises.

2. Show by sketches the sectional shading or lining used to indicate the following materials: cast iron, wrought iron, and steel.
3. Sketch the sectional shading or cross-hatching used to indicate brass, white metal, and wood.

CHAPTER VII

STUFFING BOXES, LEATHER COLLARS, ETC.

63. In cases where a reciprocating or rotating rod or spindle passes through a cylinder or casing containing a fluid, it becomes necessary to use a *stuffing box*[1] to prevent leakage of the fluid. Thus, every one is familiar with the stuffing box of a steam engine piston or valve rod, also of pump rods, and possibly, with the stuffing boxes used on the casing of a centrifugal pump where the shaft passes through it. Another interesting application that attention may be called to is the sliding expansion joint of a long steam pipe.

In Fig. 73 is shown a sketch of a **stuffing box for a horizontal rod,** lettered to show suitable proportions of its parts.[2] G is the *gland,* SS the *studs,* B the *back-bush* or *neck-brass* (which, made of brass or gun-metal, is softer than the rod, and therefore preserves the latter to a large extent from injury by wear; but when necessary the worn bush is easily replaced by a new one), SB is the *stuffing box,* and OB *oil box.*

Proportion of Parts.

$d = \frac{1}{4}\mathrm{D} + \frac{1}{4}''$ with 2 studs.	$f = 1\frac{1}{4}d$ to $1\frac{1}{2}d$.
$d = \frac{1}{5}\mathrm{D} + \frac{1}{4}''$ with 3 studs.	$g = 1\frac{1}{4}d$, or $1\frac{3}{4}d$ with oil box.
$a = \mathrm{D} + 3d + \frac{1}{4}''$	$h = a + \frac{1}{4}''$
$b = \mathrm{D} + d + \frac{1}{4}''$	$k = 2d$.
$c = \frac{3}{4}e$	$p = \mathrm{D} + 4d + \frac{1}{2}''$
$e = 5d$ to $7d$.	$j = p + 2d + \frac{1}{4}''$

It is not practicable to give any very definite rules for the proportions of stuffing boxes, as they differ with circumstances, and some parts do not vary in the same proportion as others; but the proportions given above may be taken as a guide in cases where the designer has not practical experience to fall back on.

In Fig. 74 is shown an ordinary marine type stuffing box (which has been dimensioned for a drawing exercise). The depth of the packing space S, which holds the stuffing, depends upon the pressure of the steam. In this case it has been dimensioned for a pressure up to about 150 lbs., but for a pressure of about 50 lbs. it may be made 3″ less.[3] Further, it may be explained

[1] Invented by Sir Samuel Morton.
[2] Most of these proportions are due to Unwin.
[3] Boxes that are not very get-at-able are often made very deep for the pressure they are worked at, so that they might run longer without repacking, and the same remark applies to boxes for valve rods.

that it is the practice of some engineers to make the medium-pressure and low-pressure boxes the same depth as the one for the high-pressure rod, for uniformity's sake. It will be noticed that a supplementary packing, P, is used to prevent leakage of oil and water.

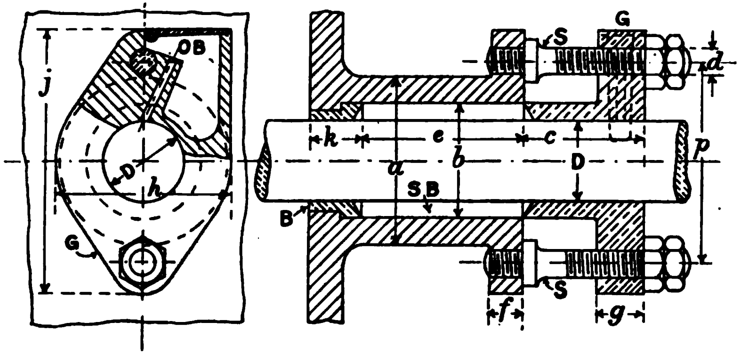


Fig. 73.—Stuffing box for a horizontal rod.

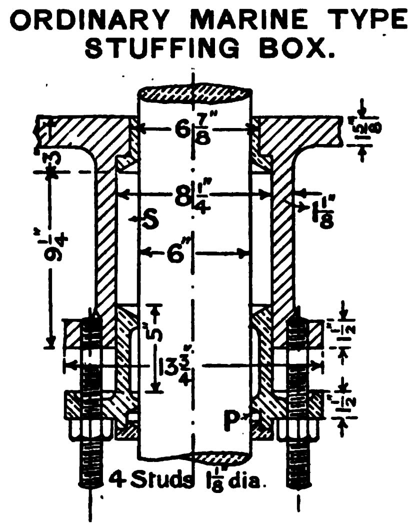


Fig. 74.

64. Stuffing Boxes with Metallic Packing.—The use of *high pressure steam* in modern practice has been the cause of a good deal of attention being given to the improvement of stuffing boxes. Although some patent vegetable and asbestos packings work very well when in good hands, with proper attention paid to ensure efficient lubrication, too often in vertical engines using high-pressure steam the lubricant only reaches the top layers of packing, keeping them soft, whilst the lower ones get hard and charred. So, for some time metallic packings have been growing in favour where high-pressure steam is used, and, although in the opinion of some engineers they have so far not been completely successful, they bid fair to entirely supersede all others, at least for the high-pressure, and perhaps even for medium-pressure, cylinders. There are a large number of different arrangements in use, differing but little in principle, so the following example should answer the student's purpose.

In Fig. 75 is shown the type of stuffing box that important piston- and slide-rods are fitted with in the **Royal Navy**, the wedge-shaped rings, GM, of gun-metal, and WM, of white metal, in contact with the rod, are made in sections scraped and fitted together, the latter being fitted with springs, S, so that any slackness due to wear in working may be taken up, to prevent movements of rings in the gland, as there is no means of adjustment when under way. Although the metallic packing effectively

bears the impact and contact of the hot steam, it generally requires to be supplemented by a few turns of soft packing—placed so that it is removed from the destructive action of the high-pressure steam—to stay any small leakage of steam which passes the metallic packing. Such an arrangement is shown at SP in this box. The neck and gland bushes are *bored large enough to allow the packing to float in the box* when slight lateral movements of the rod occur.

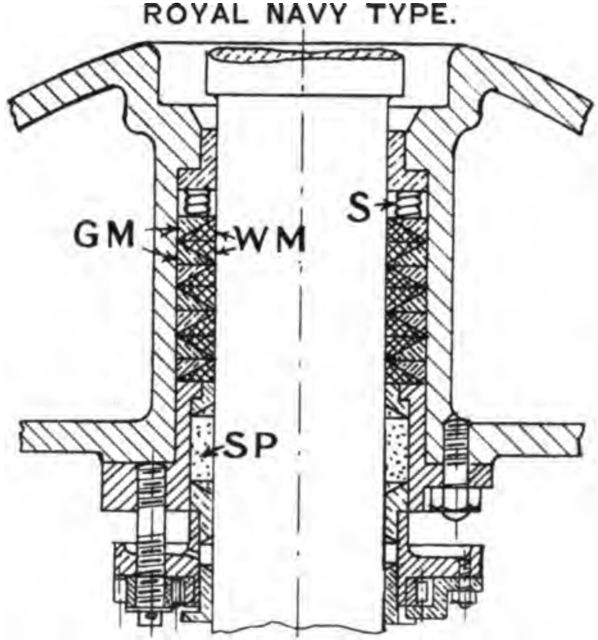


FIG. 75.—Metallic stuffing box.

65. Soft versus Metallic Packing.—No part of an engine requires more careful attention than the stuffing boxes, for if fitted with soft packing they can easily be screwed up until a very large amount of unnecessary friction is the result. They work at their highest efficiency when a faint leakage of vapour is allowed to pass out with the rod; this lubricates the packing and keeps it soft. The success in the use of metallic packing depends largely upon its being in careful hands after the designer has done his part; particularly is this the case where the pressure on the rings can be adjusted by hand. In the best practice, with excellent workmanship and experienced attendants, it gives little trouble, the wear of the piston-rods being very small. So, for high-pressure it is considered greatly superior to the old-fashioned soft packing, although the first cost is much greater.

65a. Soft Packing for Stuffing Boxes.—Any pliable packing that can be used in the packing space of an ordinary stuffing box is called *soft packing,* although the packing may be *metallic,* such as the split-rings of coiled copper or brass gauze, invented by Girdwood, for high-pressures. But the ordinary soft packing is *spun yarn* (a loose kind of hemp rope) steeped in melted tallow, and this *can only be satisfactorily used for comparatively low pressures.*

66. Leather Packing Collars.—For the rams of hydraulic machines, under certain working conditions, the most perfect packing[1] is *the* **U**-*leather,* or double cup-leather collar (invented by Benjamin Hick, of Bolton, and used by Bramah), which is shown in position in sectional elevation in Fig. 76; it is placed either in a groove in the press, as shown, or in a packing box, as shown in Fig. 77. The action is very simple, for when the water enters the press or cylinder it leaks past the part A (Fig. 77), and enters the space S (the hole in the collar is made slightly smaller than the ram, so that it is a close fit, and the outer part also

[1] Indiarubber and gutta-percha cups, also lignum vitæ and brass spring rings, have been used for packing, but on the whole, when the conditions are favourable, it is found that the leather packing cannot be improved on, particularly when it is of a close-grained flexible quality, from the middle of the back of the animal. A paper in *Cassier's Magazine* of July, 1906, stated that "leather treated with paraffin has given good results. There is no doubt that the method of preparation of the leather is an important factor in its imperviousness to water, and I have within recent years tried Vim leather, which has given better results than any I have heretofore used. The manufacturers of Vim leather claim that their peculiar process of tanning preserves the fibre, and brings the fibre into closer contact. . . . For light pressures the leather is supplied without any filler, but for high pressure the leather is filled with a lubricant which primarily hardens the leather, and renders it more impervious. . . . It is claimed that Vim leather will absorb 45 per cent. of lubricant, as compared with 15 per cent. by oak-tanned leather."

slightly larger in diameter than the groove for the same reason); the pressure water then forces one part of the collar against the ram and the other against the side of the groove, and the pressure between these parts increases with the pressure of the water, automatically making the *friction proportional to the load on the ram*, whilst with soft packing in a stuffing box the gland must be screwed down tight to prevent leakage at the highest pressure the machine is worked at, the friction being nearly the same for the smallest load. In the case of a **double-acting ram** or piston, such as is used in the cylinder of a *hydraulic crane*, two *cup leathers* (Fig. 78), are used. One of these leathers is shown in Fig. 80 in sectional elevation.

In Fig. 79 is shown a simple cup-leather arrangement which has been found to answer well in *hydraulic capstans;* and

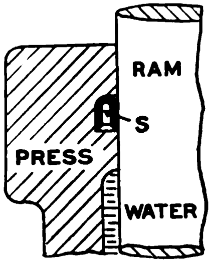


Fig. 76.—Leather collar in groove.

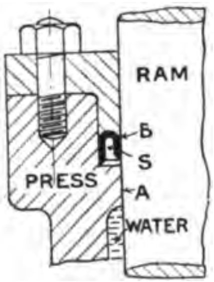


Fig. 77.—Leather collar with gland.

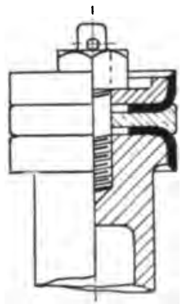


Fig. 78.—Piston fitted with cup leathers.

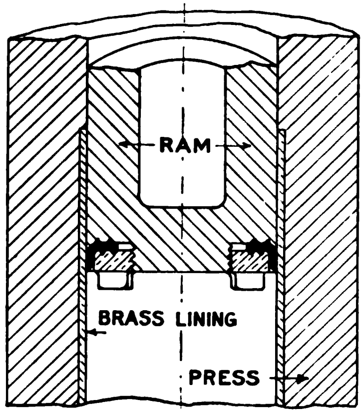


Fig. 79.—Cylinder (in section) of hydraulic capstan.


Fig. 80.—Cup leather.

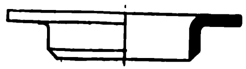


Fig. 81.—Hat leather.

in cases where the pressure water acts on one side only. The form of this collar (Fig. 81) resembles a hat,—hence its name. It is used for keeping valve spindles and rods watertight.

67. Leather Collar versus Hemp Packing.—Leather collars can be efficiently used only when they are not exposed to heat, and when they are effectively lubricated, as they soon become injured if neglected or worked in contact with water charged with gritty matter. Further, it is not practicable to use them when the ram or plunger cannot be conveniently taken out of the press to renew them. And for this reason they are not used for lift and accumulator rams, which are kept tight by ordinary hemp packing, as most pump plungers are, even where the pressures run up to 2000 lbs. per square inch, or more.

In using **hemp packing** the gasket of hemp should be *plaited very tight* and well greased, for if the plaiting be done carelessly portions are liable to be torn off when it is first used and to find their way into the valves. At first the friction with this packing is considerable, but after it has become consolidated the friction is greatly reduced. A slight leak past the packing serves to

lubricate it. But when the pressure is great and the conditions are favourable for the use of leather collars, no other kind of packing can compare with them, notwithstanding that they are more expensive.

68. Size of Leathers.—Mr. Welch, in a communication to the Institute of Mechanical Engineers, 1876, gives the thickness t for U-leather collars for great pressures as

$$t = 0{\cdot}156d^{0{\cdot}28} \quad \ldots \quad (1)$$

where d = the diameter of the ram or plunger; and for the other proportions he gives the depth D as equal to the width W (Fig. 82).

But W is more commonly made as small as practicable, and D from $1\frac{1}{2}$ W to 2 W, although, as the friction appears to be independent of the depth D, and the wear takes place near the top, it would appear that D is often made unnecessarily large. It also should not be overlooked that the deeper the collar the more the leather is strained in making it.

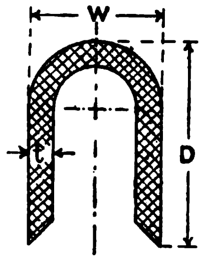


FIG. 82.—Proportions of U-leathers.

69. Friction of Leather Collars.—No very conclusive experiments on the friction of leather collars appear to have been made since those that were carried out by Mr. John Hick, of Bolton (a descendant of the inventor), many years ago,[1] from which he deduced that if—

F = the total friction of the leather collar,
P = the pressure in lbs. per square inch,
D = the diameter of ram in inches,
C = 0·0314 if collars are in good condition and well lubricated,
C = 0·047 if collars are new or badly lubricated,

Then

$$F = P \times D \times C \quad \ldots \quad (2)$$

But the total pressure on the ram $= PD^2\frac{\pi}{4}$. So let x = the fraction of ram total pressure exerted in overcoming friction then

$$xPD^2\frac{\pi}{4} = PDC$$

Hence

$$x = \frac{4C}{\pi D} \quad \ldots \quad (3)$$

EXERCISES.

DESIGN, ETC.

1. A hydraulic ram, 20″ diameter, is fitted with a U-leather collar, the water pressure being 800 lbs. per square inch. Assuming that it is badly lubricated, estimate (*a*) the total friction of the collar, (*b*) the fraction of ram pressure exerted in overcoming friction. What would you expect this fraction to be with collars in good condition and well lubricated?

[1] For an account of some interesting experiments carried out by Professor Martens, of Berlin, in which he found that the efficiency of leather collars at high pressures is nearly constant, refer to *Engineering*, Sept. 20th, 1907.

Sketching Exercises.

2. Make a freehand sketch in good proportion of a stuffing box for a 3″ horizontal piston rod.
3. Sketch a cast-iron *bushed* gland suitable for a horizontal piston rod. What is the object of making the gland of cast iron?
4. (*a*) Sketch a stuffing box for quite a small valve spindle.
   (*b*) A gland suitable for vertical stuffing box.
5. Sketch an ordinary marine type stuffing box.
6. Make a sectional sketch of the Royal Navy type of metallic stuffing box.
7. Make a bold sketch of a U-leather for a hydraulic press. Show how the collar is got into its groove if the press is not fitted with a gland.
8. Sketch a packing box fitted with a hydraulic hat leather.
9. Show by a sketch how small pump plungers are kept water-tight by leather packing.

Drawing Exercises.

10. Make half-size drawings of the stuffing box (Fig. 73) for a 4″ piston rod.
11. Make working drawings of the ordinary marine type stuffing box, shown in Fig. 74. Scale 3″ = 1′.

CHAPTER VIII

SHAFTING, CRANK SHAFTS, CRANKS, JOURNALS, ETC.

70. Shafting, its Strength, etc.—Every young engineer knows that a bar arranged to rotate about its axis, and in so doing to convey or distribute motive power, is called a *shaft*, and that shafts are supported by and rotate in what are called *bearings*, the supports of vertical shafts being either *footstep* or *collar bearings*. He also knows that the parts of the shaft which fit in the bearings are known as *journals*. And that when a shaft is too long to be made in one piece, or when it is necessary to connect two shafts or lengths of *shafting* together (ordinary mill shafting being made in lengths not exceeding 20′, and in diameters which advance by quarters of an inch), a *coupling* is used. Now the journals of a shaft are always circular in section, but the shaft or axle,[1] although its most economical section (the one that is always used by preference for the simple transmission of power) is circular, may be made square, or some other section to meet the requirements of a particular case.[2]

Now the exigencies of space forbid our going into these matters in such a way as to prove the truth of every equation used, so that the beginner without some previous training in practical mechanics, or the assistance of a teacher, could follow it; and, this being so, frequent reference will be made in foot-notes to such books as students will either possess or have the use of, so that they may plod through their difficulties if the assistance of a teacher is not available.

In calculating the diameter of a shaft the simplest cases that occur are the following:—

(*a*) **Shafts that Transmit a Uniform Twisting Moment or Torque,**[3] *and are so short that the bending action due to their own weight can be neglected.* In these it is only necessary to equate the torque (twisting moment) to the moment of resistance to twisting.

(*b*) **Shafts where Combined Torsion and Bending occur,** the bending being due to their own weight, to the thrust on a crank, or the weight of wheels, the pull of belts, etc.

71. *Case* (*a*). **Torsional Strength of a Shaft Transmitting Uniform Torque.**—In books on practical mechanics it is proved[4] that if—

d = the diameter of the shaft in inches,
T = the twisting moment or torque in lb.-inches, as measured in either of the Figs. 83, 84, and 85,
f_s = the maximum shear stress the shaft is subjected to (called skin stress),
N = Number of revolutions per minute,

[1] The term *axle* is frequently used when it would appear that *shaft* would be more appropriate, but it is not easy to lay down any fixed rule in this matter, as the terms are rather indiscriminately used, although, strictly, the term should only be used for shafts subjected to bending only.
[2] As for example, in some motor gear shafts, along which the change wheels are moved.
[3] This term is now commonly used as a synonym of *twisting moment*.
[4] In Lineham's "Mechanical Engineering," p. 417, there is a simple proof.

Then the twisting moment T = modulus of section for torsion × f_s

That is $$T = d^3 \frac{\pi}{16} f_s = \frac{d^3 f_s}{5{\cdot}1} \quad \ldots \quad (4)$$

And $$\therefore d = \sqrt[3]{\frac{5{\cdot}1\,T}{f_s}} \quad \ldots \quad (5)$$

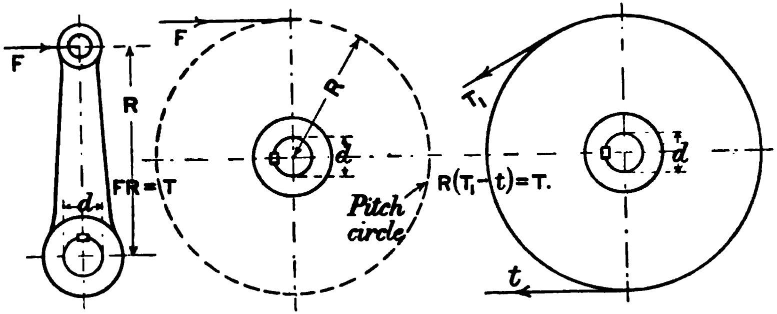
also $$f_s = \frac{5{\cdot}1\,T}{d^3} \quad \ldots \quad (6)$$

72. EXAMPLE.—A short wrought-iron shaft is to transmit a uniform torque of 20,000 lb.-ins., and the skin stress f_s has been fixed at 7500 lbs. per sq. inch. Find its diameter.

By equation (5) $$d = \sqrt[3]{\frac{5{\cdot}1 \times 20{,}000}{7500}} = 2{\cdot}386$$

Ans. $d = 2{\cdot}386''$, say $2\frac{1}{2}''$ the nearest $\frac{1}{4}''$

73. *Case* (*b*). **Combined Torsion and Bending, the Twisting and Bending Moments unvarying.**—The counter-shaft (Figs. 86 and 87) shows an example of this kind, when used to drive a machine of uniform resistance.



FIGS. 83, 84, 85.—Measurement of twisting moment T.

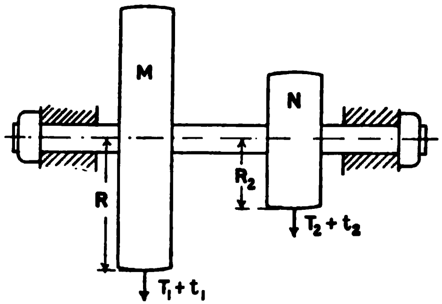


FIG. 86.

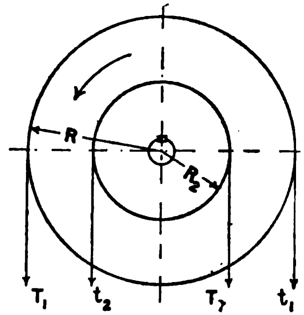


FIG. 87.

Let T = Twisting moment in inches and pounds.
„ B = Bending moment in inches and pounds.
„ T_e = An *equivalent twisting moment*, equal in its effect to *combined twisting and bending*.
„ B_e = an *equivalent bending moment*, equal in its effect to *combined twisting and bending*.

Now, most students know how to find the greatest *bending moment*, B (which of course is either at M or N), in such a case as

the above, where the positions of the pulleys in relation to the bearings and the tensions of the belts are known. Obviously, the *twisting moment*, T, equals $(T_1 - t_1)R = (T_2 - t_2)R_2$.

But before we proceed we had better be clear as to how the case stands when the—

Shaft or Axle is subjected to a Bending Moment, B, only.[1]—In this case it can be proved[2] that—

$$B = d^3 \frac{\pi}{32} f = \frac{d^3 f}{10{\cdot}2}$$

and
$$\therefore f = \frac{10{\cdot}2B}{d^3}, \text{ also, } d = \sqrt[3]{\frac{10{\cdot}2B}{f}} \quad \ldots \ldots \quad (7)$$

where f equals the skin stress in tension and compression per sq. inch.

74. EXAMPLE.—A wrought-iron revolving axle is subjected to a bending moment B = 25,000 lb.-ins., and the maximum stress is 5000 lbs. per sq. inch. Determine its diameter.

By equation (7)
$$d = \sqrt[3]{\frac{10{\cdot}2 \times 25{,}000}{5000}} = 3{\cdot}7$$

$$\textit{Ans. } d = 3{\cdot}7'', \text{ say } 3\tfrac{3}{4}''$$

75. But **when the shaft is subjected to combined twisting and bending**, it is proved in works on the theory of elasticity[3] that—

$$T_e = B + \sqrt{B^2 + T^2} \quad \ldots \ldots \quad (8)$$

and
$$B_e = \frac{B + \sqrt{B^2 + T^2}}{2} \quad \ldots \ldots \quad (9)$$

Thus
$$T_e = 2B_e$$

So, equating the moment of resistance to twisting (Eq. 4), to the equivalent twisting[4] moment (Eq. 8), we have—

$$\frac{d^3 f}{5{\cdot}1} = B + \sqrt{B^2 + T^2}$$

[1] Some axles are fixed and the wheels rotate on them (the back axle of a chain-driven motor-car, for instance); obviously they are subjected to bending only.

[2] Lineham's "Mechanical Engineering," p. 430.

[3] A very simple proof of this, Rankine's, formula is also given in Goodman's "Mechanics applied to Engineering," p. 489, and in Innes' "Problems in Machine Design," p. 74. The formulæ are only true for *circular* sections.

[4] The **equivalent twisting moment** (also **equivalent bending moment**) is only equivalent to the actual combined moment, in the sense that they produce the same greatest direct or shearing stress. The above formulæ are due to **Rankine**, who argued that the **second skin stress**, *i.e.* the one perpendicular to the maximum stress, had no effect. But many experiments go to show that his formulæ, if strictly used, should only be applied to **brittle materials**. However, usually the factor of safety employed is such that there is a sufficiency of strength, particularly as there is a large reserve of metal that is very little stressed. For these reasons, coupled with the fact that the formula is in general use, Rankine's has been made use of in working the examples which follow:—

The **French formula** for the *equivalent bending moment* is $B_e = \frac{3}{8}B + \frac{5}{8}\sqrt{B^2 + T^2}$, but this appears to be almost equally unsatisfactory, with the further disadvantage that, it is not so easily manipulated. However, the **Guest formula** (Art. 76) is more simple in form, more easily handled, and it agrees more closely with experimental results for **ductile materials** such as wrought iron and mild steel.

or,
$$d = \sqrt[3]{\frac{(B + \sqrt{B^2 + T^2})5{\cdot}1}{f_s}} \quad \ldots \quad (10)$$

And if $B = FL$ and $T = FR$, a more convenient form is—
$$d = \sqrt[3]{\frac{F(L + \sqrt{L^2 + R^2})5{\cdot}1}{f_s}} \quad \ldots \quad (10\text{A})$$

75a. EXAMPLE.—The diameter of a mild steel crank shaft is to be fixed, the radius of the crank being 16″, the overhang (distance between centre of crank pin and centre of bearing) 10″, the maximum thrust on crank pin 10,000 lbs., and the greatest skin shear stress, 6000 lbs. per sq. inch.

By equation (10a)
$$d = \sqrt[3]{\frac{10{,}000(10 + \sqrt{10^2 + 16^2})5{\cdot}1}{6000}} = 6{\cdot}25$$

Ans. $d = 6{\cdot}25''$

76. Guest's Equivalent Bending Moment.—We have seen (Art. 75) that the Rankine and French formulæ usually employed for the equivalent bending moment, are somewhat inconvenient in form for expeditious use, and, strangely enough, it has been found that for the ductile materials generally used for cranks and cranked parts, the results do not very well agree with experiments that have been made.

In this connection it should be explained that Mr. James J. Guest communicated a valuable paper to the Physical Society, which was read May 25th, 1900 (see *Phil. Mag.*, July, 1900 [1]), in which he shows that, in the case of crank shafts, the **greatest stress and greatest strain theories** lead to too small dimensions, and he shows that the **greatest shearing stress theory** leads to the **equivalent bending** moment B_e having the following value—

$$B_e = \sqrt{B^2 + T^2}$$

And from experiments that have been made, this formula (known as the **Guest**) agrees very closely with the results for **ductile metals**. It has the added advantage of being very simple, and, of course, when applied, allows of a lower factor of safety being used than should be employed for such ductile metals as wrought iron and mild steel with either the Rankine or French formula.

The Guest formula easily lends itself to **graphic treatment**. For, referring to Fig. 88, we have $B = FL$, and $T = FR$, where F is the thrust on the crank pin in a direction at right angles to the crank. Then (as in Art. 75) we have

$$B_e = \sqrt{B^2 + T^2} = F\sqrt{L^2 + R^2} \quad \ldots \quad (11)$$

But AD, the hypotenuse of the right-angled triangle ACD,[2] equals $\sqrt{L^2 + R}$, and by construction, $AE = AD$. So that F acting on the end E of the arm AE subjects the crank shaft at A to a bending moment which is equivalent to the actual combined bending and twisting moment, in the sense that it produces the same greatest direct or shearing stress.

[1] Refer also to *Phil. Mag.*, Feb. and Oct., 1906, and to results of tests by Mr. E. S. Hancock, American Society of Testing Materials, 1905 and 1906. Also to Perry's "Applied Mechanics," p. 356.
[2] Euc. I. 47.

So that in cases where the **diameter of the shaft** has to be determined we have, by equating the moment of resistance to bending to B_e, the equation [1]—

$$d^3 \frac{\pi}{32} f = F\sqrt{L^2 + R^2}$$

or

$$d = \sqrt[3]{\frac{10{\cdot}2F\sqrt{L^2 + R^2}}{f}} \quad . \quad . \quad (11\text{A})$$

where f is the skin stress in tension and compression.

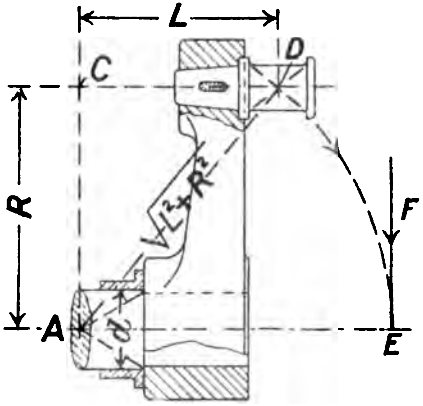


FIG. 88.

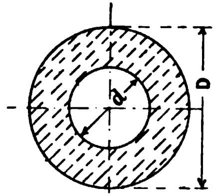


FIG. 88A.—Hollow shaft.

77. Hollow Shafts.—Shafts are made hollow to distribute the material more efficiently and to reduce their weight. Obviously, the strength of the hollow shaft (Fig. 88A) is the strength of a solid shaft of diameter D less the strength of a shaft the size of the hole d, **as it would be stressed if it formed part of the larger one.**

So, for hollow shafts **the torsional strength** $\mathbf{T} = \frac{(D^4 - d^4)}{D} \frac{\pi}{16} f_s$

Let [2] $d = xD$

Then

$$T = \frac{D^4 - (xD)^4}{D} \frac{\pi}{16} f_s$$

or

$$T = D^3(1 - x^4) \frac{\pi}{16} f_s$$

whence

$$D = \sqrt[3]{\frac{T\ 5{\cdot}1}{(1 - x^4) f_s}} \quad . \quad . \quad . \quad (12)$$

78. Hollow and Solid Shafts of Equal Strength.—In the case where a hollow shaft is to replace a solid one (of the same material) their moments of resistance to twisting must be equal, or

$$d_1^3 \frac{\pi}{16} f_s = \frac{(D^4 - d^4)}{D} \frac{\pi}{16} f_s$$

That is (Art. 77),

$$d_1^3 = \frac{D^4 - d^4}{D} = D^3(1 - x^4)$$

Therefore

$$D = \sqrt[3]{\frac{d_1^3}{1 - x^4}} \quad . \quad . \quad . \quad (13)$$

[1] Of course the equivalent twisting moment $T_e = 2F(L^2 + R^2)$, and this would be equated to the moment of resistance to twisting $d^3 \frac{\pi}{16} f_s$.

[2] The steel shafts in naval ships are invariably made so that they may have the greatest strength with the least weight. The general practice is to make $x = 0{\cdot}5$, or $\frac{d}{D} = \frac{1}{2}$.

And if $$x = \frac{1}{2}, \text{ then } d_1^3 = D^3\frac{15}{16}$$

or $$D = d_1\sqrt[3]{\frac{16}{15}} = 1{\cdot}022d$$

And thus we find that for the hollow shaft to equal in strength the solid one it would only be slightly over 2 per cent. larger, and its weight would be 22 per cent. less than the solid one.

As a further compensation for the increase in the cost of its production, it is much stiffer against bending by its own weight than the solid one, and therefore the bearings can be further apart. It is usual to close the mouth of the bore after erection to prevent corrosive action due to water getting inside and between the flanges of the couplings.

79. Example.—It has been decided to replace a solid 16″ shaft by a hollow one made of steel 10 per cent. stronger. The hole to be 0·6 the outer diameter. Find the size of the hollow shaft, and the ratio of the weights of the two shafts, assuming that their densities are the same.

In this case the shafts are made of different materials, so

$$d_1^3 f_s = D^3(1 - x^4)f_s', \text{ or } d_1^3 = D^3(1 - x^4)\frac{f_s'}{f_s} \qquad (14)$$

Then $$D = \sqrt[3]{\frac{d_1^3 f_s}{(1 - x^4)f_s'}} = \sqrt[3]{\frac{16^3 \times 100}{[1 - (\frac{6}{10})^4]110}} = 16{\cdot}24''$$

And the ratio of weights $$= \frac{D^2 - d^2}{16^2} = \frac{162{\cdot}4^2 - 8{\cdot}12^2}{16^2} = \frac{66}{100}$$

Ans. D = 16·24″, say $16\frac{1}{4}$″ Ratio of wts. = 66 to 100

Representing a saving in weight of 44 per cent.

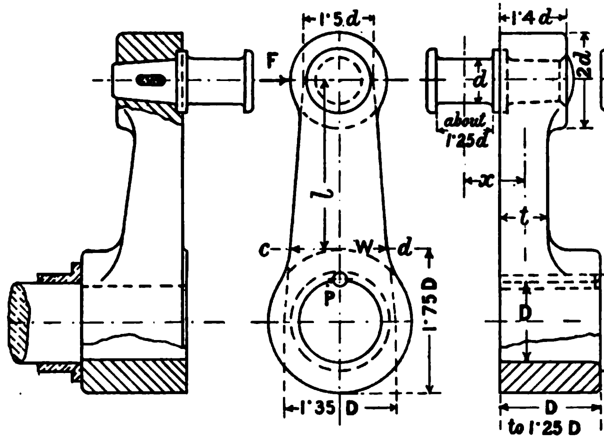
80. Cranks.—In Articles 75A and 76 we have discussed some matters relating to crank shafts. It now remains to give attention to *cranks* and *cranked shafts*, but although the engineer is called upon to deal with a number of types, only the more important ones can be explained in these pages. The best shape for a crank depends partly upon whether it is made of cast iron, wrought iron, or steel. By far the most reliable material is wrought iron or steel, the last named being now almost exclusively used; indeed, cast iron is rarely used now except for small cranks for cheap machinery. But the expense of forging large cranks is very great, more particularly as it is impossible to forge them very near the required form, which means that the weight in the rough, and cost, are greatly in excess of that due to the finished size. And these are the reasons why cast iron is occasionally used for cranks; when they are well proportioned and the work is not too severe, they rarely fail. On the other hand, for such jobs as deep well pumps, where the work is apt to be very trying, they are not to be recommended; and, further, at the best they must be larger than wrought-iron ones, which is often a disadvantage.

In Figs. 89 and 90 are shown two views of an *overhung crank* of the usual form for wrought iron or steel. The proportions marked on them may be used for ordinary cases.

In the best practice the boss of the crank is either shrunk on or forced on by hydraulic pressure; a key then is not required, but a small pin is driven in a hole drilled partly in the boss and shaft as shown (Fig. 90), a flat or groove being made on the pin to allow the air to escape when it is driven in. The *crank-pin* is usually fixed to its boss by riveting, as shown in Fig. 91, alternative fixings being shown in Figs. 89 and 91A, the latter also showing an alternative way of finishing the end of the crank shaft where the crank is fitted.

81. Built-up Cranks.—The marked increase in recent years of the dimensions of crank shafts for marine purposes has led to the old-fashioned wrought-iron shafts being almost entirely replaced by steel ones. Although at first these occasionally failed after running some time, breaking at the junction of either the shaft or crank-pin with the web, this trouble has to a large extent been overcome by using softer steels, and by careful annealing after being forged. Larger fillets and more rigid bearings in more perfect alignment have also helped to reduce the number of failures.

Such shafts have the additional advantage of smaller pins and journals, with lighter connecting rods, and in adopting them we have got rid of the uncertainty which was always felt as to the soundness and continuity of wrought iron in heavy built-up forgings.



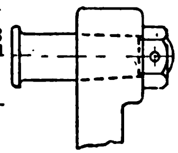
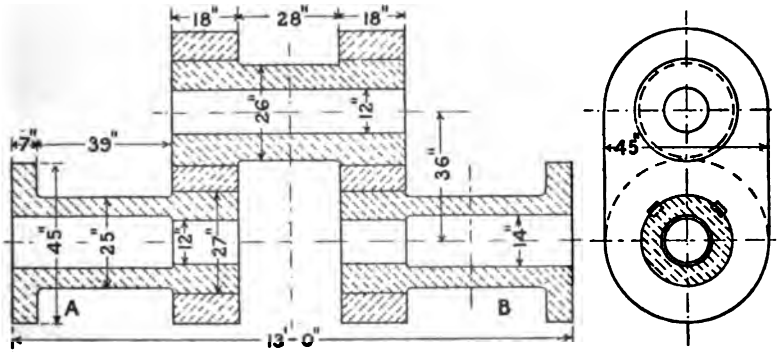


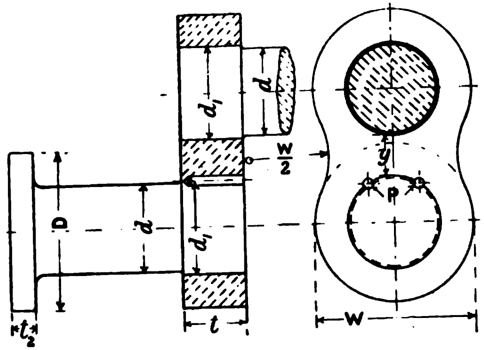
FIG. 91A.

FIG. 89. FIG. 90. FIG. 91.

Cranks of forged steel or wrought iron.



FIGS. 92, 93.—Built-up fluid-compressed steel cranks.

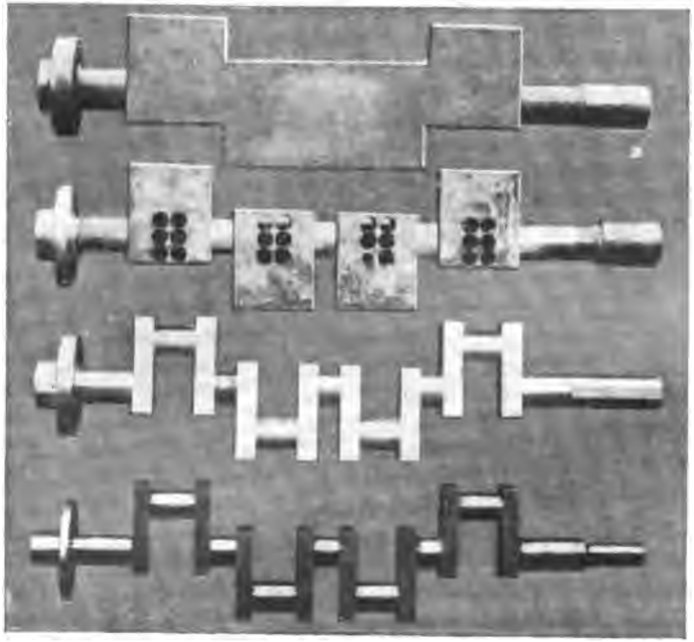


FIGS. 94, 95.—Built-up crank. Ordinary type.

To avoid the whole shaft being scrapped should one part fail, shafts above 10″ diameter are generally made in duplicated halves or in three lengths the same size, so that a *spare length* need only be carried for use in the event of a breakdown.

For diameters over 15″ *built-up shafts* have been commonly used for some years, the ordinary practice being to rough-machine the parts, fasten the shafts and pins into the flat slabs which form the webs, and then turn the shaft all over and finish as though it were a solid one. But in *building up* the finest work much greater care is taken; thus in Figs. 92 and 93 are shown a fine example of what is probably at present the most perfect practice in the construction of marine crank shafts. It is one of the three cranks made at Sir Joseph Whitworth's works at Manchester, from fluid-compressed steel, for the S.S. *City of Rome*, its weight being over 20 tons.

The shafts A and B were forged hollow, and the flanges afterwards forged out to the proper diameter. They were then rough-bored and turned all over, the parts fitting into the webs being finished-turned, $\frac{1}{20}$″ diameter being allowed on the other parts for turning and polishing; after the parts had been shrunk together, the webs were keyed as well as shrunk[1] on to the shaft. The pins were forged hollow, rough-bored, and turned to size, then oil-hardened and ground up true in the lathe by emery wheels, being thus completely finished before being shrunk into the webs. The webs were forged from very large ingots into slabs, the ends punched and worked all round the eye on a mandril. They were then planed, bored together in pairs, and the ends were shaped on a slotting machine.



1
2
3
4

FIG. 96.—Four stages in the manufacture of a petrol motor crank shaft.
1. Crank shaft rough forged, planed on flat, and collar rough turned.
2. Crank shaft webs formed externally and drilled.
3. Crank shaft rough turned and oil tempered.
4. Crank shaft finished.

In Figs. 94 and 95 are shown the form most favoured for large cranks, where weight is not the first consideration.[2] The usual proportions are $W = 1{\cdot}8$ to $2d$, $t = 0{\cdot}6$ to $0{\cdot}75d$, $d_1 =$ about $\frac{41}{40}d$, $t_2 = 0{\cdot}25$ to $0{\cdot}28d$. The distance y should not be less than $0{\cdot}45d$, or there would be danger of the pin and shaft working loose. The webs or cheeks are either shrunk on or forced on to the shaft and pin; in either case the hole is made $\frac{1}{1000}$ of the diameter less than the shaft or pin to be fitted. If a single key

[1] $\frac{1}{1000}$ of the diameter was allowed for shrinkage, this amount being determined as most suitable by trial with specimen pieces; the force required to push one part from the other having been previously observed.

[2] This type is nearly always used in merchant vessels, where weight is not the most important consideration, but they are seldom found in war-ships.

pin is used, its diameter may be $\frac{1}{15}d. + 0{\cdot}39''$, but two smaller ones, as shown at P, are better, as they do not weaken the part between the pin and shaft.

82. Petrol Motor Crank Shafts.—The stages in the manufacture of a slotted-out **motor-car crank shaft** are shown in Fig. 96 (taken by kind permission from Messrs. Willans and Robinson's pamphlet), but the operations vary little in the manufacture of this type of

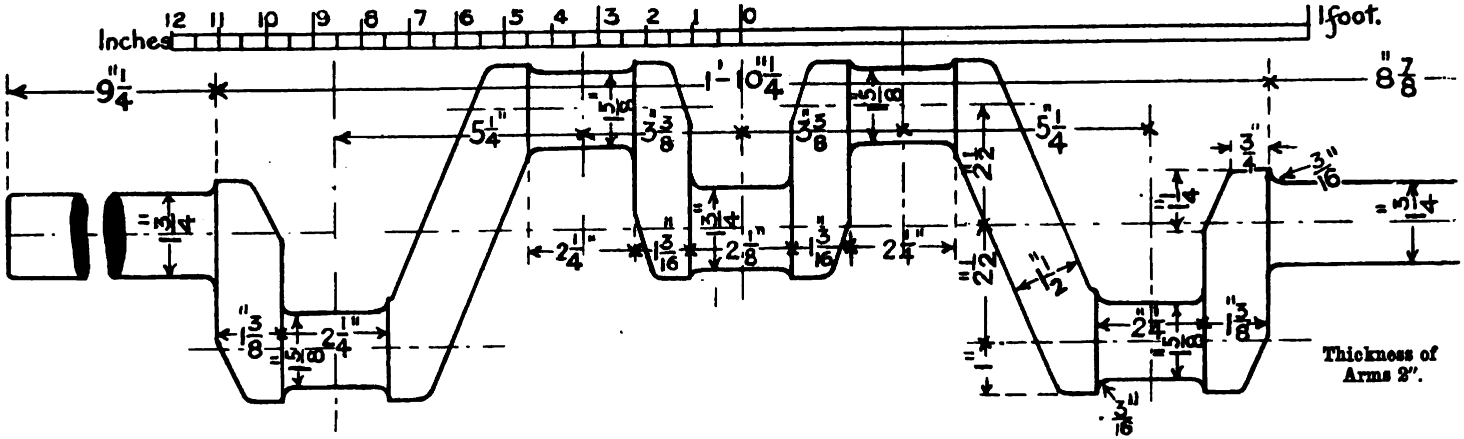


FIG. 96A.—Petrol motor crank shaft. Offset type.

shaft, whether it be for a marine or motor car. The drawing should speak for itself. The greatest care need be exercised in selecting material for crank shafts, but this particularly applies to motor-car work, where the dimensions must be kept down, and, therefore, in the best practice no expense is spared to secure material of the very highest quality.

Experience has proved that one of the finest steels for this purpose, if not the finest, is **oil-tempered vanadium steel**, which shows a unique combination of *high static strength* (and especially a high *yield-point*) with high resistance to *shock and fatigue;* indeed, it is claimed for it that it is the best steel yet produced for dealing with the severe strains[1] to which crank shafts are subject.

83. Petrol Motor Crank Shaft (Drawing Exercise).—We have shown in Fig. 96A a petrol motor crank shaft of the **offset** type. It is fully dimensioned and may be drawn to a **scale of half full size.**

[1] The crank shafts of internal combustion engines during the suction stroke of the piston are subject to a reversed torque.

EXERCISES.

Design, etc.

1. A shaft of 15" diameter is fitted with a crank of 40" radius in such a way that the bending action may be neglected. The maximum thrust on the crank end is 149,000 lbs. Find the skin stress of the shaft. *Ans.* 9000 lbs. per sq. inch.

2. A steel shaft of a crane transmits a torque due to a force of 2 tons acting in the pitch circle (36" diameter) of a wheel fixed to it, and the maximum shear stress is 9500 lbs. per sq. inch. Find the diameter of the shaft. Note.—Crane shafts, being very short, are usually designed for strength alone, and not torsional stiffness. The wheel being close to the journal, neglect effect of bending action. *Ans.* $d = 3{\cdot}51$, say 3½".

3. The driving pulley of a machine is 30" diameter, and the effective tension of the belt is 150 lbs. What horse-power is given to the machine when the pulley makes 120 revolutions per minute? *Ans.* H.P. = 4¾.

4. The upright shaft of a turbine transmits 30 H.P. at 250 revolutions per minute, and the skin stress is 10,000 lbs. per sq. inch. Find the diameter of the shaft. *Ans.* $d = 1{\cdot}56''$, say 1¾".

5. The trunnions of a mixing machine have an effective length of 10", and the weight which comes on each one is 1½ tons. What should their diameter be if the skin stress is not to exceed 5500 lbs. per sq. inch? Note.—In this arrangement you are to assume that the trunnions are only subjected to bending. *Ans.* $d = 3{\cdot}145''$, say $d = 3\frac{1}{4}''$.

Drawing Exercises.

6. Make a set of drawings of the built-up crank (Figs. 92 and 93). Scale 1" = 1'.

Sketching Exercises.

7. Make a sketch of a steel overhung crank, and explain how it may be fixed to the shaft. Also show two ways of fitting the crank pin. Which do you prefer, and why?

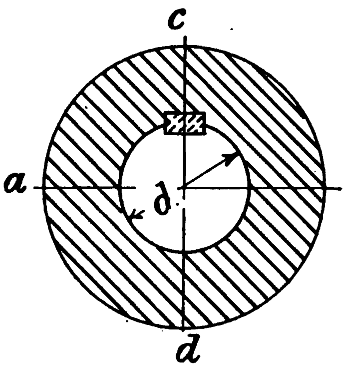
CHAPTER IX

COUPLINGS, CLUTCHES, ETC.

84. We have explained that whenever a line of shafting exceeds some 20′ in length it is made up of two or more lengths, connected together by what are technically called *couplings*, many forms of which are in use. One of the simplest of these is the *butt-muff coupling*, three views of which are shown in Figs. 97, 98, and 99. They are arranged to form a drawing exercise, in continuation of the previous ones, and it will be convenient to touch upon the principal features of the arrangement as we describe how it may be drawn.

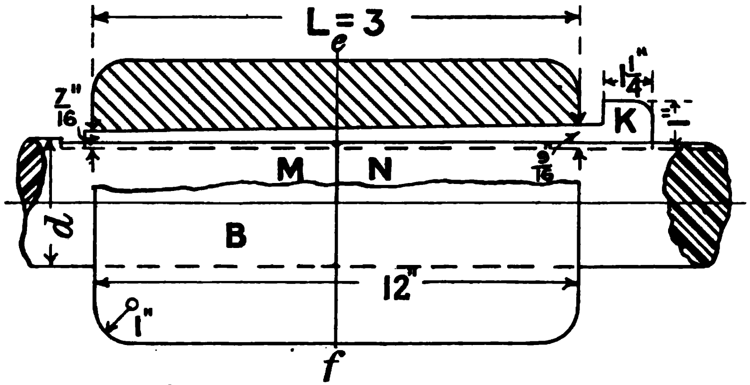
85. Drawing Exercise.—To draw a Butt-muff Coupling. (Scale, half full size.)—From an inspection of the figures[1] it will be

BUTT-MUFF COUPLING.



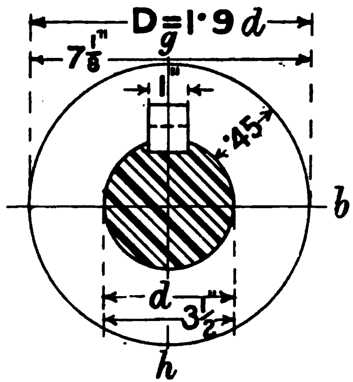
SECTION *ef*.

Fig. 97.



SECTIONAL ELEVATION.

Fig. 98.



END ELEVATION.

Fig. 99.

[1] It will be seen that the proportional parts in terms of the *unit* are given, but it has been also dimensioned for a 3″ shaft as a drawing exercise.

seen that the *sleeve muff*, or *box*, B, is slid over the ends M and N of the two pieces of shafting that butt, and are required to be coupled together, and a taper key, K, is used, as shown, to fix the box to the shafting so that one length may transmit a torque or twisting action to the other. Now, remembering what we have said about commencing a drawing of an object that has a circular part, it will be seen that this is a case where the end views, Figs. 97 and 99 (or as much of them as possible), should be drawn first; so, having drawn the circles, the **section of the key** (taken through the centre of the coupling), Fig. 97, can be set out. As this is an important detail, it is shown in Fig. 100 to a larger scale. The point A, on the centre line and circle, is the centre of the section, and the thickness of the key here should be half its breadth. Now, the rule for **breadth** W is, $\frac{d}{4}+\frac{1''}{8}$, then in this case, $W=\frac{3\frac{1}{2}}{4}+\frac{1''}{8}=1''$, and $BC=\frac{1}{2}''$, so *the* **depth**, AC, **of the keyway** (which is uniform throughout the length[1]) becomes $\frac{1}{4}''$, the full taper of $\frac{1}{8}''$ to the foot being given to the keyway in the box. Fig. 101 shows the key in pictorial projection.

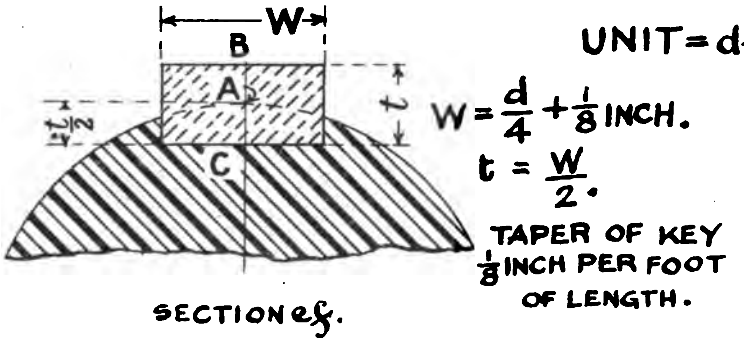


FIG. 100.—Proportions of sunk key.

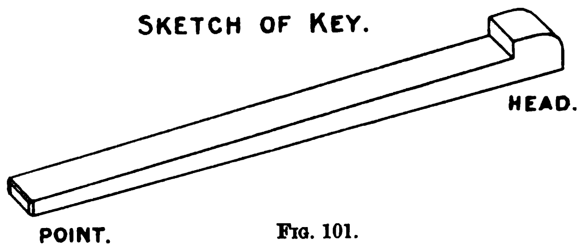


FIG. 101.

With these hints, the student should now experience no difficulty in drawing the three views shown, and in setting out a complete plan of the coupling. He will notice that he is instructed to make the drawings to a scale of *half full size*, that is to say, he is to draw the object one-half its real size, but he would not *dimension* the drawing with figures one-half of the original ones, as the **dimensions on a drawing** *indicate the real size*, and are independent of the scale to which the drawing may be made. All horizontal dimensions are placed to read horizontally in the spaces left for them between the dimension lines, and all vertical dimensions read from bottom to top of drawing when looking from its right-hand edge. The points of the arrow-heads must touch the lines between which the dimension is taken.

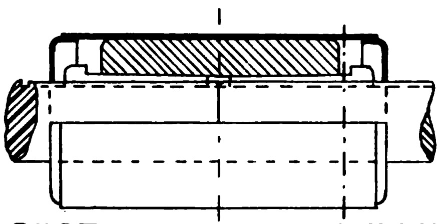
Every important part should be dimensioned on at least one of the views, and in cases where a body consists of two or

[1] The taper is always made on the coupling or boss, which is fitted to the shaft, excepting when the *key* is fixed, and the boss moves along the shaft a short distance; the key (which is then called a *feather*) is then parallel. Refer to Chapter X.

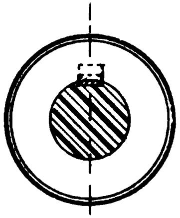
more divisions of its length, breadth, or thickness, the *overall* (sum of its parts) dimensions should be shown; indeed, in some cases it saves time in reading a drawing (when it gets into the works) if important dimensions are occasionally repeated on different views.

It should be explained that although the muff-coupling is the simplest one in general use,[1] it requires to be very carefully fitted if it is to be a first-rate job, for, obviously, unless the depth of the keyway in each of the shafts to be coupled be exactly the same and the diameters be the same, the key will be bedded on one shaft whilst the other will be loose. To prevent this happening, some engineers make the **key in two lengths**, and drive them both in from the same end, one for each shaft. Or they

CASED BUTT.

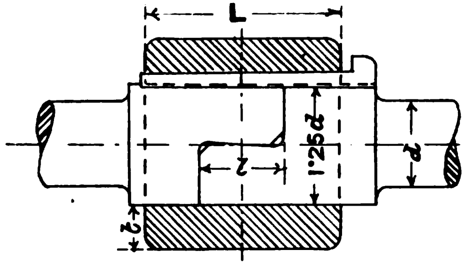


SECTIONAL ELEVATION.
FIG. 102.

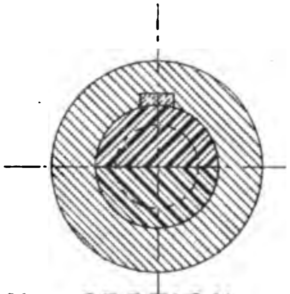


END ELEVATION.
FIG. 103.

LAP BOX.



SECTIONAL ELEVATION.
FIG. 104.



SECTION.
FIG. 105.

may be driven from opposite ends, as shown in Fig. 102. This figure and Fig. 103 also show how the **coupling** is **cased** when it is necessary to protect the key-heads from coming into contact with the clothes of workers.

Proportions.—Taking the *unit* as $d + \frac{1}{2}''$, the usual proportions are shown on the figures in terms of the unit.

Materials.—The box is made of *cast iron;* the shafts, usually of wrought iron or mild steel; and keys, of mild steel.

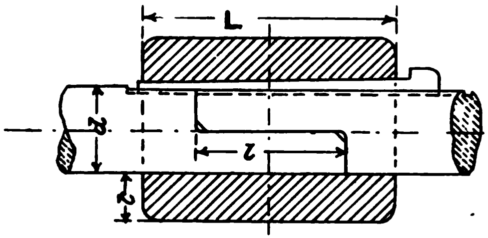
86. Fairbairn's Lap-Box Coupling.—Figs. 104 and 105 show an excellent but expensive coupling introduced by the late Sir W. Fairbairn, but not often used now. The usual proportions are, in terms of unit $= d + \frac{1}{2}''$, $L = 2$, $l = 0{\cdot}8$, $t = 0{\cdot}45$, taper of lap about 1 in 12. The function of the key in this case is only to prevent the box sliding off the joint, so a *saddle* key is used, as shown; but, as the shaft is weakened where the lap is formed, this part is made 25 per cent. larger than the diameter of the shaft.

A cheaper form (but a weaker one) of this coupling is shown in Figs. 106 and 107, the ends not being enlarged, which is allowable in some cases, as it does not often happen in line-shafting that the shaft is strained to its full *torsional* strength. With unit $= d + \frac{1}{2}''$ the proportions may be $L = 2{\cdot}6$, $l = 1{\cdot}5$, $t = 0{\cdot}45$. A sunk key is shown, but one fitted on a *flat*,[2] or even a saddle key, would suffice.

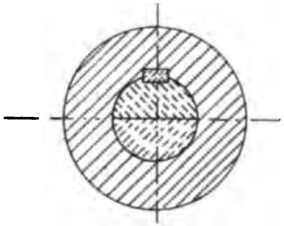
[1] These box couplings can be used as pulleys when required, which is an advantage.
[2] Seated on a flat, machined or filed on the shaft, the same breadth as the key.

87. Flange Couplings are largely used, particularly for shafts over 3″ diameter. Figs. 108 and 109 show the form this coupling takes when it has to transmit a torque equal to the full strength of the shaft (which does not very frequently happen); the ends of the shafts are then enlarged as shown, so that they are not weakened by the keyways. For a line of shafting to be in perfect alignment, the greatest care is necessary in fitting the couplings, so in the best practice they are either shrunk on or forced on by hydraulic pressure; about $\frac{3}{8}$″ length of the shaft being arranged to project from one face to enter the other and keep the two in position.[1] The keys, with these tight fits, need not be fitted very tightly, but when the flanges are not shrunk or forced on, it is most important that they should be carefully fitted top and bottom, and well driven home, or they will work loose; this slightly cants the coupling, and to correct this the length of shafting is put in the lathe and the flanges trued up after keying. Either gib-headed keys, as in Fig. 108, or keys driven from the face, Fig. 110, are used. The latter makes the safest[2] job.

HALF LAP.



SECTIONAL ELEVATION.
FIG. 106.



SECTION.
FIG. 107.

The following proportions are those given by Unwin, with slight modifications:—

PROPORTIONS OF FLANGE COUPLING (Figs. 108 and 109).

Let N = number of bolts (refer to following Art.).
δ = diameter of bolts.
d = diameter of shaft.

Then

Unit $= d + \frac{1}{2}''$	$d = 1{\cdot}12$	$l = 1{\cdot}25$
$D = 2 + 6{\cdot}5\delta$	$t = 0{\cdot}5$	$\delta = \dfrac{0{\cdot}6d}{\sqrt{N}}$
$D_1 = 2$	$x = 2{\cdot}5\delta$	

The screwed part of the bolt may $= 0{\cdot}7\delta$ in diameter, and the minimum number of bolts is 4; for shaft over $1\frac{1}{2}$″ diameter N may equal $\frac{1}{2}d + 3$. It is usual to use only even numbers of them, but there is no rational reason why this be so, if the coupling is properly fitted.

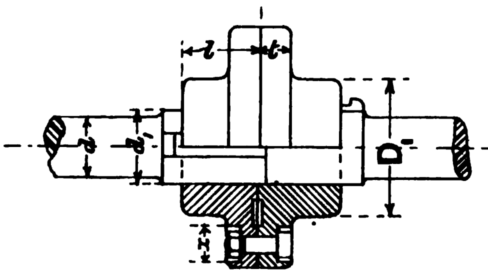
88. Strength of Bolts.—In ordinary line shafting the bolts in a flange coupling are often subjected to a good deal of tensional

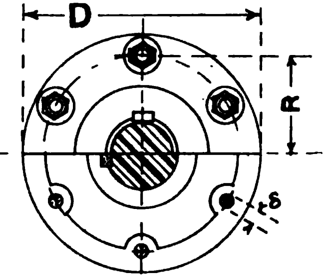
[1] Of course this can only be done when the shafts are the same diameter; in cases where they are unequal, the *faces of the coupling* are made spigot and socket to preserve alignment, but when the shafts are long, this arrangement is often very troublesome if a length is to be dismounted, as the faces cannot be slid on one another.

[2] Projecting keys are very dangerous, and should never be used in exposed positions unless protected. The author witnessed a terrible fatal accident due to such heads.

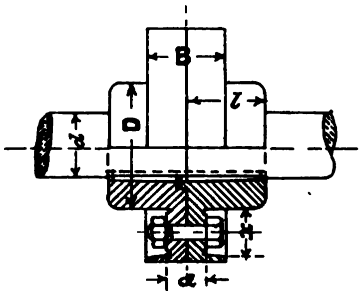
stress due to the bending action of the shaft, and as this is in addition to the pure shear stress the bolts resist due to the torque, this latter stress must be proportionally reduced. From an examination of several cases that have stood well in practice, and

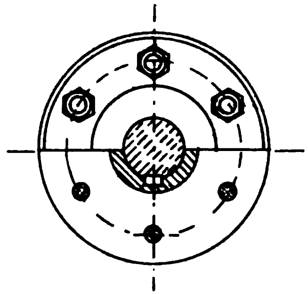
FLANGE COUPLING.





PULLEY FLANGE COUPLING.





SECTIONAL ELEVATIONS.

FIG. 108. FIG. 109.

SECTIONAL ELEVATIONS.

FIG. 110. FIG. 111.

appeared to be well designed, the shear stress in the bolts had a mean value of about $\frac{4}{9}$ the skin stress of the shaft. This being so, we will deduce an equation that will enable us to readily determine the bolt diameter from the usual data.

So, assuming $f_s = 9000$ for shaft, and $f_s = 4000$ for bolts, the shear strength of one bolt × R × N = moment of resistance to twisting of shaft.

$$\therefore \delta^2 \times \frac{\pi}{4} \times 4000 \times R \times N = d^3 \frac{\pi}{16} \times 9000$$

And
$$\delta^2 = \frac{d^3 \times 9}{4 \times 4 \times N \times R}. \quad \text{But on an average } R = 1{\cdot}5d$$

$$\therefore \delta = \frac{0{\cdot}6d}{\sqrt{N}} \text{ nearly} \qquad (15)$$

And this value we have used in the proportions above. Of course, should δ come out an odd fraction, the diameter of bolt used would be the nearest $\frac{1}{8}''$ above.

89. Pulley Flange Coupling.—Figs. 110 and 111 show a slightly modified and useful form of the coupling just examined. This is fitted to shafting without enlarged ends, which are expensive to make and have the disadvantage that split pulleys have to be used. It can be used as a small pulley when convenient.

SUITABLE PROPORTIONS.

$\delta = \dfrac{0{\cdot}6d}{\sqrt{N}}$	$l = 1{\cdot}4$
Unit $= d + \frac{1}{2}''$	$a = 0{\cdot}55$
$D = d + 0{\cdot}8$	$B = 0{\cdot}55 + 3\delta$

90. Claw Coupling.—Many years ago Mr. Box called attention to the evil effects of the rigidity of muff and flange couplings, more particularly when used for large shafts, say of 6″ diameter or more, as there is want of accommodation in case of bad adjustment in fixing or faulty alignment, or further, should a bearing be neglected and wear down considerably more than the rest. With small shafts this is not so serious, as they spring, and more or less adjust themselves to such irregularities, but a heavy shaft may be too stiff to do so, and therefore requires a yielding coupling, such as the one shown in Figs. 112 and 113, which consists of two castings (one on each shaft) each with three projections or claws which fit in corresponding recesses in the opposite casting. When fitted or machined in the usual way these couplings become rather expensive, but Mr. Box suggested a very simple way of inexpensively making them which has much to recommend it. He proposed to cast one half *upon the other*, by first casting one part in the usual way and then using that as a *chill* to cast the projecting parts of the other on, the chilled surfaces lasting longer, as an additional advantage. When made in this way, the castings would be locked together as they come from the mould. They are bored, then separated, the keyways cut, and they are ready for use. Our drawings show the more usual form of this coupling as arranged for disengaging, the part A being keyed to the shaft, and the other part B being made with a groove G into which the forked end of a lever works, so that by a movement of it the part B can be slid along the *feather* F (a parallel key fixed to the shaft) engaging or disengaging the sliding part with the other part, and in so doing connecting or disconnecting the shafts. The former, of course, can only be done when the driving shaft is moving very *slowly;* even then a great strain is thrown upon the shaft and its fittings.

CLAW COUPLING.

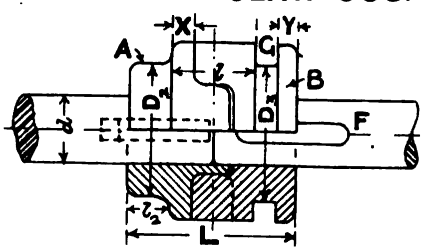


FIG. 112.

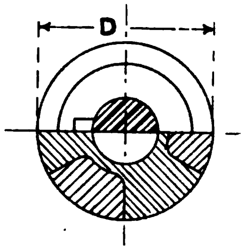


FIG. 113.

PROPORTIONS OF CLAW COUPLING.

Unit $= d + 1$	$D_2 = 1{\cdot}6$
$L = 2{\cdot}65$	$l_2 = 0{\cdot}6$
$D = 2{\cdot}1$	$X = Y = 0{\cdot}3$
$D_3 = 1{\cdot}5$	$l = 1{\cdot}25$

91. Propeller and Crank Shaft Coupling (Figs. 114 and 115).—These shafts, whether they be solid or hollow, are connected

by flange couplings, the flange being forged with the shaft. Conical bolts are now generally used to hold the flanges together and transmit the torque. The taper is generally made from 1 in 25 to 1 in 15, with the larger taper the heads being often dispensed with. The bolt holes must not only be accurately bored but also rimered out to the correct taper when the flanges are together. The screwed part of bolts is made smaller than the body, as shown in the figures., to keep down the size of the nuts and flange.

We have seen (Art. 88) that when d = the diameter of a shaft and δ the diameter of the bolts, N the number of bolts and R the radius of the bolt circle,

$$\delta^2 \frac{\pi}{4} f_s' \mathrm{RN} = d^3 \frac{\pi}{16} f_s$$

or for hollow shafts

$$\text{,,} \quad = \left(\frac{\mathrm{D}^4 - d^4}{\mathrm{D}}\right) \frac{\pi}{16} f_s$$

but the bolts are generally the same material as the shaft, and if we assume that they are subjected to shear stress only, then $f_s' = f_s$ and R may $= 0{\cdot}7d$ (or 0·7D, the outer diameter of *hollow* shafts).

Then
$$\delta = \frac{d}{2}\sqrt{\frac{1}{0{\cdot}7\mathrm{N}}} \quad \ldots \ldots \quad (16)$$

which varies from $\frac{d}{6}$ to $\frac{d}{4}$, according to the number[1] of bolts used. This number being rarely less than $\mathrm{N} = \frac{d}{2}$. So the following proportions may be used:—

92. Proportions of Marine Coupling (Figs. 114 and 115).—Thickness of flange $= 0{\cdot}25$ to $0{\cdot}28d$ (or 0·25 to 0 28D for hollow shafts).

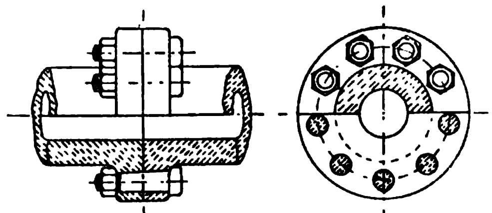
Minimum number of bolts, $\mathrm{N} = \frac{d}{2}$ or $\frac{\mathrm{D}}{2}$ for hollow shafts

Diameter of bolt circle $= \mathrm{D}' = 2\mathrm{R} = 1{\cdot}4d$ or 1·4D for hollow shafts

Mean diameter of bolts
$$\delta = \frac{d}{2}\sqrt{\frac{1}{0{\cdot}7\mathrm{N}}}$$

and for equal strengths of solid and hollow shafts
$$d^3 = \frac{\mathrm{D}^4 - d^4}{\mathrm{D}}$$

MARINE TAIL SHAFT.



SECTIONAL ELEVATIONS

FIG. 114. FIG. 115.

In naval practice a larger number of smaller bolts are used to keep the diameter of the flanges down. The diameter of flange equalling $1{\cdot}4d + 2{\cdot}2\delta$ or $1{\cdot}4\mathrm{D} + 2{\cdot}2\delta$.

[1] An even number of bolts is used for a *two-crank* engine, whose shaft is in duplicate halves; whilst for a *three-crank* engine, whose shaft is in three duplicate parts, the number of bolts is a multiple of 3.

EXERCISES.

Design, etc.

1. The diameter of the bolt circle of a flanged coupling for a 5″ shaft is 15″, and there are 6 bolts. What should their diameter be if their working stress is 8000 lbs. per sq. inch, the skin shear stress of the shaft being 9000 lbs. per sq. inch? *Ans.* $d = 1\frac{1}{16}$″.

Drawing Exercises.

2. Make a set of drawings of a butt-muff coupling for a 4″ shaft. Scale, $4\frac{1}{2}$″ = 1′.
3. Make a set of working drawings of a flange coupling for a 5″ shaft. Scale, quarter full size.
4. Make working drawings of a cast-iron claw coupling for a 3″ shaft. Scale, half full size.
5. Draw three views of a marine crank shaft coupling; diameter of shaft 12″. Scale, $1\frac{1}{2}$″ = 1′. (For proportions, refer to Art. 92.)

Sketching Exercises.

6. Sketch a butt-muff coupling. And also show an alternative way of fixing with the key in two parts. Why is the latter arrangement likely to make the best job?
7. Sketch a cased butt coupling using two gib-head keys.
8. Sketch a cheap form of lap coupling.
9. Sketch a pulley flange coupling. What advantage has this form over the ordinary one shown in Figs. 108 and 109?
10. Sketch a claw coupling. What precautions must be taken in engaging this coupling? What factor limits its use?
11. Sketch a marine tail shaft coupling. Why are the bolts made taper?

CHAPTER X

KEYS AND PIN KEYS, ETC

93. We have seen that when two parts, such as a coupling (or wheel) and a shaft upon which it fits, are to be fixed to prevent a rotary motion of one about the other, they are generally *keyed* together. Now, there are various ways of doing this, which vary considerably in relative costliness and efficiency, and an examination of the following examples should enable the young engineer to select the one which will on the whole answer his purpose best in any given case.

A **key** is a wedge with parallel sides. When both ends are accessible, it is made without a head, as it can be drifted[1] out. A side view of this form is shown in Fig. 116. In cases where the small end of the key is inaccessible, the key is made with a head, as in Fig. 117, and becomes a **gib-head key**; a wedge may then be forced between the key head and boss of wheel or coupling to withdraw the key.

Taper.—The taper found to answer best is $\frac{1}{8}''$ to the foot of length.

94. Saddle or Hollow Key.—This key, with the usual proportions, is shown in Figs. 118 and 119. It is only suitable for light work, as rotatory slip is prevented by friction alone.[2]

TYPES OF KEYS AND KEYING.

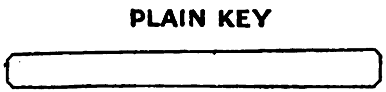


Fig. 116.

Fig. 117.

SADDLE OR HOLLOW KEY.

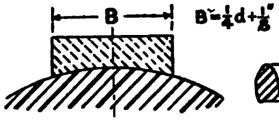


Fig. 118.

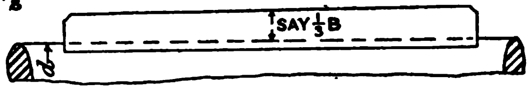


Fig. 119.

95. Key on Flat.—Figs 120 and 121 show this key. The flat on the shaft being parallel to the axis, the draught or taper is made on the part to be attached to the shaft, as in the previous case. This arrangement is more secure than the preceding one, and is suitable for rather heavier work.

96. Sunk Key.—This key, which is shown in Figs. 122 and 123, is always used for heavy work; the key in this case fits in a *sunk keyway*, whose bed is parallel to the axis of the shaft, and whose depth is half the mean thickness of the key, measured

[1] A tool resembling a bent blunt chisel, used for forcing out keys with a hammer, is called a **key-drift**.
[2] Pulleys or riggers fixed in this way can easily be shifted to another position on the shaft.

from the top of the shaft, as shown in Fig. 100. This key requires very skilful fitting; the keyway in both parts should be exactly the same breadth, and the key should accurately fit the sides, whilst theoretically it should touch the top and bottom with a light pressure only, to avoid straining the boss. But every practical man knows that, unless it is carefully fitted, with

KEY ON FLAT.

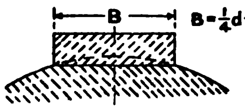


FIG. 120.

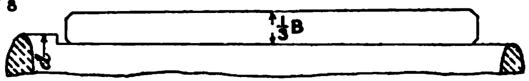


FIG. 121.

SUNK KEY.

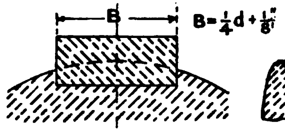


FIG. 122.

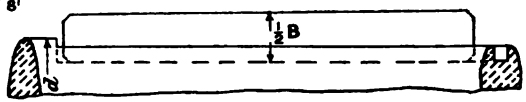


FIG. 123.

the whole of the top and bottom well bedded, and it is forced in with a driving fit, such that a blow will make the whole thing ring, it will sooner or later work loose, which, under some conditions, might be a serious matter, and under any conditions is objectionable. The tendency to work loose is greatly increased when the torque is irregular.

97. Two Keys.—In cases where the hole is a shade larger than the shaft, or in reversing jobs, it is advisable to use *two keys, fixed at right angles* to each other; one of these is usually either a saddle key, or a key on the flat, whose function is to cause the shaft to be gripped in three places, and so prevent rocking on the shaft.

98. Key Boss.—In heavy work the weakening effect of the keyway in the boss of a wheel cannot be neglected, so the thickness, t, of the boss is maintained, or slightly increased, as shown in Fig. 124, by arranging what is called a *key boss,* AB. The drawing should speak for itself.

99. Staking On.—In Figs. 125 and 126 are shown two examples of what is called staking on. Should the solid boss of a wheel have to be passed over an enlarged end of a shaft, it is usual to fit it to the shaft by using four keys bedded on flats on the shaft,[1] as in Fig. 125. Where the strains are very great, and the shaft is square, Fig. 126 shows the most reliable way of fixing a wheel. Four temporary keys are first fitted in the spaces, K, and the wheel truly centred, after which the permanent keys are accurately fitted. Only those parts on which they bed need be machined. The keys are numbered and marked, so that the job can easily be re-erected.

100. Cone Keys.—For light work, where a frictional drive is practicable, instead of staking a wheel on a shaft, as explained in the previous article, cone keys may be used. The three keys are cast in one piece, with wrought-iron dividing plates; they can be bored to the size of the shaft, and turned with the usual key taper to fit the hole in the pulley boss, before parting them into the three pieces, which, after trimming, are ready for use as saddle keys, as shown in Figs. 127 and 128.

101. Pins.—For light work it is often convenient to use **taper pins,** instead of ordinary keys. Thus, in Figs. 129 and 130, the boss of a lever is shown pinned on to a shaft or spindle. The holes, after drilling (for all such pins), should be rimered out to a total taper of $\frac{1}{4}''$ in $1'$, and the mean diameter of the pin, d, may be $\frac{1}{4}$ D. In Fig. 131 a small pulley or hand-wheel is

[1] For light work, cone keys are used in this way. (Art. 100.)

shown fixed this way, but in this case the hole can only be drilled on the slant, as shown. When the materials of the boss and shaft are the same, or very nearly alike in character and hardness, they can be drilled for the reception of a cylindrical pin, as

KEY BOSS.

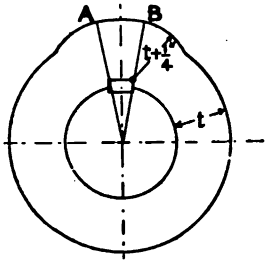


FIG. 124.

STAKING ON.

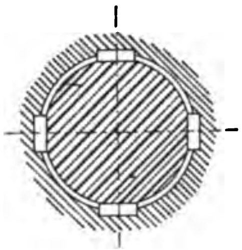


FIG. 125.

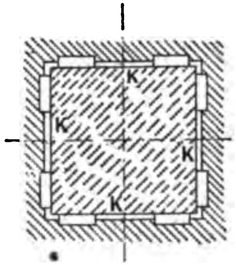


FIG. 126.

CONE KEYS.

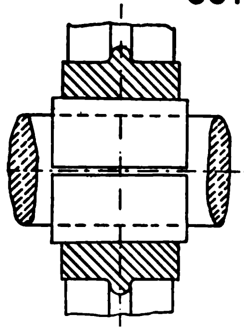


FIG. 127.

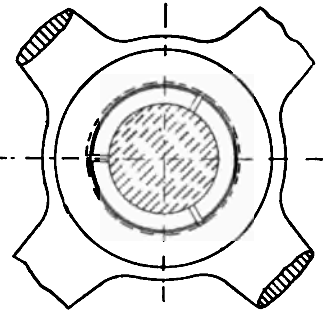


FIG. 128.

in Fig. 132; the diameter, *d*, of the pin may be $\frac{1}{6}$ D to $\frac{1}{4}$ D, according to the length of the boss. In Figs. 133 and 134 are shown two lengths of pipe, connected by taper pins and a dowel. This is a joint which is largely used for railway signalling

APPLICATIONS OF TAPER PINS.

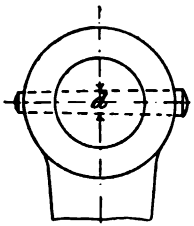


FIG. 129.

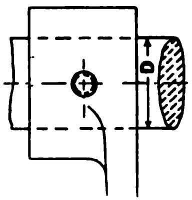


FIG. 130.

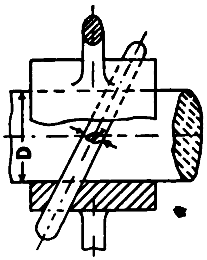


FIG. 131.

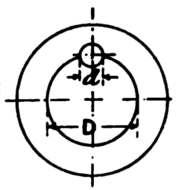


FIG. 132.

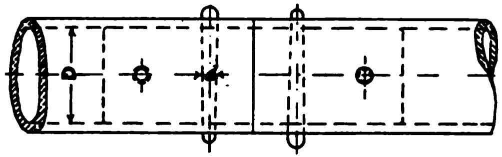


FIG. 133.

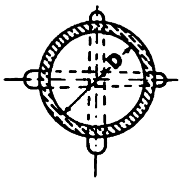


FIG. 134.

rods, which are alternately in tension and compression, also for ventilating machinery, the rods transmitting a twisting action.

The following Table 1, gives the standard dimensions for taper pins:—

TABLE 1.—Standard Taper Pins.

Taper ¼″ per foot.

No. of pin.	Total length of pin.	Largest diameter of pin.	Smallest diameter of pin.	No. of pin.	Total length of pin.	Largest diameter of pin.	Smallest diameter of pin.
	Inches.	Part of an inch.	Part of an inch.		Inches.	Part of an inch.	Part of an inch.
0	1	0·156	0·135	6	3·25	0·341	0·279
1	1·25	0·172	0·146	7	3·75	0·409	0·331
2	1·5	0·193	0·162	8	4·5	0·492	0·398
3	1·75	0·219	0·183	9	5·25	0·591	0·482
4	2	0·250	0·208	10	6	0·706	0·581
5	2·25	0·289	0·240				

102. Feathers.—When a wheel or some part of a machine is to be secured to a shaft, in such a way that it must rotate with it, but is free to be moved in the direction of the axis of the shaft, a *feather* is used. Now, this feather or sliding key is a parallel strip which is usually fixed to the shaft, the groove in the wheel boss or sliding piece being made a working fit. Or, alternatively, the feather is fixed to the sliding piece, and slides in the feather way of the shaft. Figs. 135 and 136 show one arrangement of the former, the feather being very slightly dovetailed where it fits the shaft, and the metal of the shaft each side of it being lightly caulked down to secure it.

Figs. 137 and 138 show an alternative way of fixing the feather by two countersunk screws, as at A (one only shown), or by forging on the feather two pins to pass through the shaft with countersunk riveted heads, as at B (one only shown).

VARIOUS FORMS OF FEATHERS.

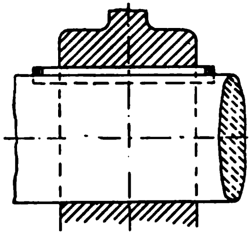


Fig. 135.

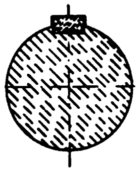


Fig. 136.

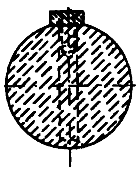


Fig. 137.

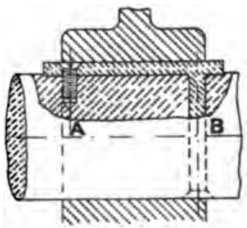


Fig. 138.

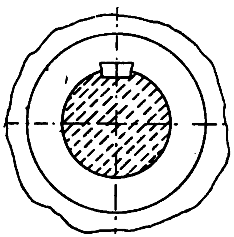


Fig. 139.

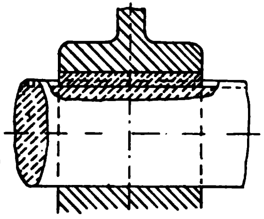


Fig. 140.

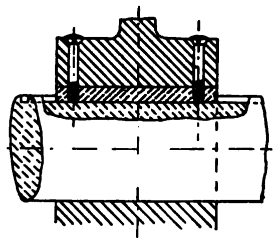


Fig. 141.

Figs. 139 and 140 show the feather fixed to the boss by dovetailing, the feather being a driving fit in the boss, whilst Fig. 141 shows how it is fixed by two screws through the boss. Sometimes by forging two pins on the feather it is fixed to the

boss by riveting, in a similar way to that shown at B, Fig. 138. Other alternative arrangements are shown in Figs. 142, 143, and 144. In Fig. 142 the feather is made with a lug, F, which is attached to the side of the boss by means of a screw. To make a job of this the workmanship must be very accurate. Figs. 143 and 144 show loose feathers fitted to the bosses; in the former the feather is made with a projecting pin, which is placed in the hole to prevent any end movement, and the same purpose is served by the gib-heads of the feather in the latter. In both of these the shaft must admit of the boss and feather being passed over its end.

VARIOUS FORMS OF FEATHERS.

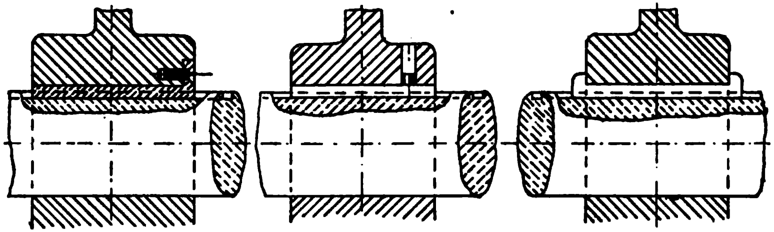


FIG. 142. FIG. 143. FIG. 144.

103. Strength of Keys.—When keys are made in accordance with the empirical proportions[1] marked on Figs. 118 to 123, they very rarely give trouble or fail unless they are made exceptionally short in proportion to their other dimensions. Nevertheless, as cases occur (particularly with large crank shafts) where the part secured by the key takes off only a small proportion of the power transmitted by the shaft, it will be instructive to show how they should be treated, with the help of Fig. 145.

Now, let L = length of the key.

B = breadth „ „

t = mean thickness $= \frac{B}{2}$

f_s = safe shear stress of shaft per square inch.

f_s' = safe shear stress of key per square inch.

f_c = safe compressional stress of key and shaft per square inch, say $= 2f_s$.

F = safe shear force acting on key $= LBf_s'$.

F = also safe load in compression on sides $= L\frac{1}{2}tf_c$.

T = moment of resistance to twisting of shaft $= d^3\frac{\pi}{16}f_s$ = also $\frac{Fd}{2}$

Then, if the crushing resistance of the key is to be equal to its shearing resistance, $L\frac{1}{2}tf_c = LBf_s'$, that is $Bf_s = \frac{1}{2}tf_c$.

But f_c may equal $2 \times f_s'$, therefore, the key to be equally strong to resist crushing and shearing would have to be square, which is about its section

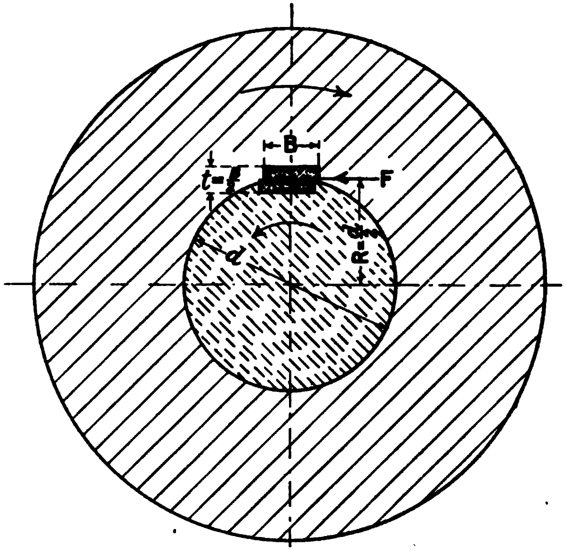


FIG. 145.—Shearing resistance of keys.

[1] The following proportions of *sunk keys* are recommended by Box: where D = diameter of shaft, B = the breadth of key, T = thickness of key, d = depth sunk in the shaft measured at the side of the key, then B = (D ÷ 4) + 0·125, T = (D ÷ 11) + 0·16, and d = (D ÷ 40) + 0·075.

when it takes the form of a *feather*, and is not a tight fit top and bottom. But for ordinary *keys* we have seen that $B = 2t$, and, if the full strength of the shaft is to be transmitted by the key, we must not overlook the wedge action, which considerably relieves the sides of the crushing effect. Hence it is usual in designing a special *feather* to take into account the *crushing* effect, but in dealing with a *key* to take into account its *shearing* resistance. So, let us assume that we wish to determine the length L of the key shown in Fig. 145, in terms of the other quantities, where t = mean thickness and the materials of the key and shaft are the same, that is $f_s = f_s'$. Then for equal[1] strengths of shaft and key—

$$d^3\frac{\pi}{16}f_s = \frac{Fd}{2} = LBf_s'\frac{d}{2}, \text{ and } \therefore L = \frac{d^2\pi f_s}{8Bf_s'}$$

Or, when the shaft and key are of the same material, we have $L = \dfrac{d^2\pi}{8B}$ . . . . (17)

104. Example.—The full strength of a 3″ steel shaft is to be transmitted through a steel key. Find the dimension of the key.
From the empirical proportions, Fig. 122 and 123—

$$B = \tfrac{3}{4} + \tfrac{1}{8}'' = \tfrac{7}{8}'' \text{ and } t = \frac{B}{2} = \frac{1}{2} \times \frac{7}{8} = \frac{7}{16}$$

Then by (Eq. 17)—

$$L = \frac{d^2\pi}{8B} = \frac{3^2 \times 22}{8 \times \frac{7}{8} \times 7} = 4{\cdot}04$$

Ans. L = say, 4″

EXERCISES.

Design, etc.

1. The coupling of a 4″ steel shaft has to transmit the full strength of the shaft. What should be the dimensions of a steel key for this purpose?
2. A 12″ lever is fixed to a 1½″ shaft by means of a taper pin passed through its boss, as in Figs. 129 and 130, the mean diameter of the pin being ⅜″. What pull on the end of the lever would cause a shear stress on the pin of 9000 lbs. per sq. inch? and what skin stress in the shaft would this correspond to?
3. A treadle lever 30″ long is fitted to a break shaft 1¼″ diameter, the maximum load on a treadle being 200 lbs. A key $\frac{7}{16}$″ wide is used to fix the lever to the shaft. What should the length of the key (and that of the lever boss) be, if the shear stress is not to exceed 8000 lbs. per sq. inch?
4. A lever is fixed to a 2″ shaft by a taper pin, as in Figs. 129 and 130; the mean diameter of the pin is ½″. Find the ratio of the shearing stress of the pin to the skin stress of the shaft.
5. A lever is fixed to a 3″ shaft by a taper pin, as in Figs. 129 and 130, and the skin stress of the shaft is the same as the shear stress on the pins. What must be the mean diameter of the pin?

[1] After looking through the above, the case where these strengths are *unequal* should be easily worked. Obviously the strength of the key is proportional to its length.

6. An inch pipe has an outside diameter of $1\frac{5}{16}''$, and it is arranged as a shaft, with a lever fixed to it, as in Figs. 129 and 130. What should the size of the pin be if its shear stress be 50 per cent. greater than the skin stress of the pipe?

Sketching Exercises.

7. Show by sketches three different ways of keying wheels to a shaft, and explain under what conditions each would be used in practice.
8. Show by a sketch how the boss of an important wheel is strengthened where the keyway is cut by a key boss.
9. Make sketches showing how wheels are staked on to round and square shafts. Under what conditions is *staking on* necessary?
10. What are the conditions which allow a wheel to be fixed to shaft by *cone keys*? Make a sketch of the arrangement.
11. Levers, hand-wheels, and small pulleys are sometimes *pinned* on to a shaft or spindle. Show two ways of doing this.
12. Lengths of metal tubing are sometimes connected by a dowel and taper pins. Show how this is done, and mention any application of this joint you are acquainted with.
13. What is a feather? In what important respect does it differ from a key? Sketch three or four characteristic examples of *feathers*.

CHAPTER XI

RIVETED JOINTS

105. ONE of the most simple and efficient fastenings, which has been extensively used for a great variety of purposes from very ancient times, is the rivet. As a fastening, it somewhat resembles a bolt, but differs from it in two important respects; for a *bolt can be used as a temporary fastening*, and can be withdrawn by unscrewing the nut; but a *rivet is a permanent fastening* and the parts held together by it can only be separated by chipping off a head. Further, a bolt is used satisfactorily when the straining force acts in the direction of its axis, giving it a tensional load, but it is not considered safe to load a rivet in this way, its proper function being to resist shearing in a direction normal to its axis.

Rivets are made in special machines, from special round iron or steel bar,[1] with heads either **cup-shaped**, as in Fig. 146, or **pan-shaped**, as in Fig. 147, formed while red hot by dies of these shapes, and their finished forms before use (showing the length of rivet required to form the head) are shown by the dotted lines.

In **riveting plates**, whenever practicable, *riveting machines* are used;—the rivet is made red hot, passed through the plates and pressed between two dies by hydraulic or steam pressure. The heads are then usually made **cup or spherical shaped**, as in Fig. 146, and are said to be **machine riveted.** When machines are not available, the rivets are **hand riveted.** For this job a full gang consists of three men and a boy, the latter to heat the rivet and bring it from the furnace to the holder up, who inserts it into the rivet hole and presses against the rivet with a tool called a **dolly**, cupped to receive the head of the rivet, while the other two men on the opposite side hammer the other end down with riveting hammers and finish it off by a blow or two from a sledge hammer, a **snap-headed** tool being interposed to give the head the cup shape in Fig. 146. In confined positions where it is not possible to snap the heads, they are finished by hammering[2] to the conical or *conoidal* form shown in Fig. 148, which has not quite the strength of the cup head. In many classes of work, such as the skin of ships,[3] the seatings of girders, etc., the heads must not project; the plates are then countersunk, as shown in Fig. 147 (which shows a *full* counter-sunk head), and the heads finished off flush with the plate, or with a slight fulness or projection, as shown dotted. Fig. 149 shows a form that is sometimes given to this head to prevent its sharp edge springing away from the plate. Fig. 151 shows the half-countersunk head. Fig. 150A shows how, by slightly countersinking the holes, the head can be a little strengthened; it is usual to somewhat reduce the size of the heads, as shown, when this is

[1] See Art. 106.

[2] Rivets up to 1" or $1\frac{1}{4}$" in diameter may readily be closed with hammers of 8 to 10 lbs. weight; but if the head is to be formed in a die or swage, a heavier hammer, say 16 lbs. weight, is necessary. Skilful riveters on bridge work can rivet up from 200 to 250 per day when the size is $\frac{5}{8}$", and from 90 to 100 if 1". On vertical members about 75 per cent. of these may be done, and for boiler work about twice as many.

[3] Lloyds' Registry have fixed the size and shape of rivet heads for this purpose for diameters from $\frac{3}{8}$" to $1\frac{3}{8}$".

done. For drawing purposes an approximation to the ordinary cup head is easily made by using a radius of $\frac{3}{4}$ the diameter, as shown in Fig. 150A, and striking the head from a point on the centre line $\frac{1}{8}$D from the shoulder.

106. Proportions of Rivet Heads, etc.—These proportions vary somewhat in practice, as they have not yet been standardized,[1] but those shown on Figs. 146 to 150 may be taken to be average ones; they are in terms of D, the diameter of the hole. The dotted lengths for forming the heads should be taken to be approximate. They vary from 1·25 to 1·7 times the diameter, the actual length required depending upon the completeness with which the rivet fills the hole, and upon whether the head is formed by hand or by machine; the former requires about $\frac{1}{8}$D less length than the latter, as the machine compresses and swells the rivet till it completely fills the hole, thus making a very perfect job.[2]

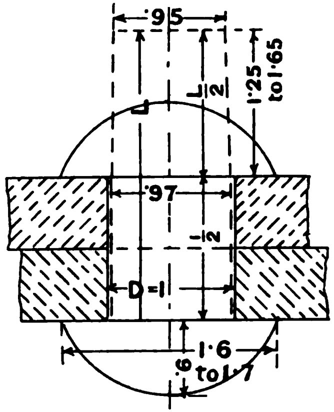
Great care must be taken in dealing with long rivets, as when they are some 6

[1] In some firms it is the practice to make the rivet diameter for sizes above $\frac{7}{8}$" smaller than the holes by $\frac{1}{16}$".

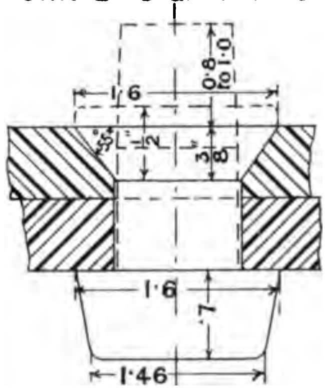
[2] Machine work has been for some years largely superseding hand riveting. The machines perform their work much more rapidly and economically. They were first used for bridge work on the famous Conway Tubular Bridge. Rivets can only be made to solidly fill the holes by freeing them from the oxide and slag, which can be done by heating them to a bright red singly, and passing them through a fine spray of water, the chilling action causing the slag and oxide to shell off, leaving a perfectly clean rivet.

PROPORTIONS OF RIVETS, ETC.

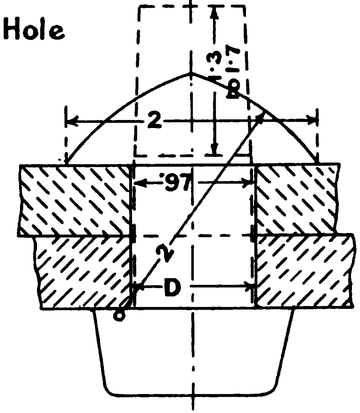
Unit D - Diam. of Rivet Hole



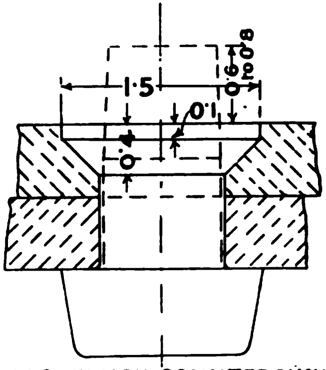
146. CUP OR SNAP HEAD.



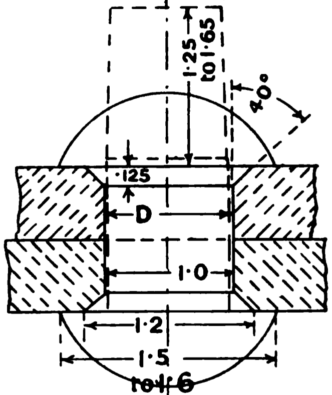
147. FULL COUNTERSUNK HEAD.



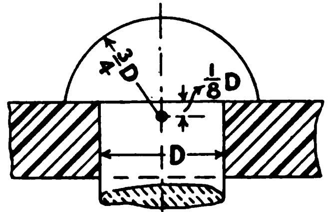
148. CONOIDAL HEAD HAMMER FINISHED.



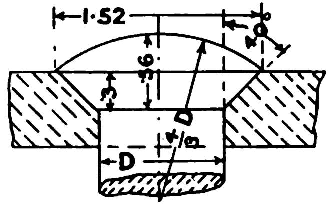
149. FLUSH COUNTERSUNK



150. SNAP COUNTERSUNK.



150A. PROPORTIONS FOR DRAWING PURPOSES.



151. HALF COUNTERSUNK HEAD.

to 8 diameters in length, they often contract enough to draw off their heads, so, to avoid this in very long rivets, the head end should be cooled before placing the rivet in its hole.

107. Rivet Materials, etc.—With iron plates, soft ductile iron of a strong, tough, good quality, with a tensile strength not exceeding 54,000 lbs., and giving an elongation of not less than 25 per cent. in 8″ is used for the rivets. Formerly such iron for rivets was largely used with steel plates, but there is now no difficulty in getting a soft low-carbon steel suitable for rivets, with a tensile strength not greater than 54,000 lbs. per square inch, and an elongation in 8″ of 30 per cent., and such rivets are generally used now with steel plates. And for **boiler** purposes the Board of Trade is satisfied with **steel plates** of an ultimate tensile strength f_t of 28 tons per square inch, with an ultimate shear strength f_s of 23 tons per square inch for **rivet steel**.

108. Drilling and punching Rivet Holes.—Many years ago the general practice was to punch all rivet holes for girder, bridge, and boiler work, with the result that the spacing and alignment of the holes were very imperfect,[1] and this want of accuracy became more pronounced when two plates, each with a row of punched holes, were brought together to be riveted, and led to the objectionable practice of more or less **forcing the holes** into agreement by hammering a conical drift into them; even then a fairly large proportion of the rivets would be forced into position in such a way that their sectional area was materially reduced at the joints just where the shear occurred.

But the use of multiple-drilling machines and high-speed tool steel has enabled engineers in increasing numbers to considerably reduce this evil by drilling the plates, and it looks as though the barbarous practice just referred to, if not punching generally, will be almost or entirely superseded. Certainly, in the best boiler work all the holes are drilled, and most of them when the plates are together in position. It should be further explained that when plates are punched, particularly steel ones, the metal round the hole is injured by the lateral flow of the metal under the pressure of the punch. This injury, however, may be entirely removed in either of two ways, for if the plate is annealed after punching it is restored to its original condition, or, if the hole is punched $\frac{1}{16}$″ smaller than is required and rimered or drilled out to size, the injured material is removed[2] and the plate is ready for use.

109. Caulking and Fullering.—Joints in boilers, tanks, etc., are made fluid-tight by *caulking*. Fig. 152 shows how this is ordinarily done; T being a narrow, blunt chisel-like tool, called a *caulking tool*, about $\frac{3}{16}$″ thick at the end and $1\frac{1}{2}$″ in breadth, the edge ground to an angle of 80°. It is moved after each blow along the edge of the plate, which is usually planed to a bevel of about 75° to 80°, to facilitate the forcing down of the edge. It will be seen that the tool burrs down the plate at B, forming a metal-to-metal joint, care being taken *not to damage the plate*[3] below the tool, or spring the joint open. Usually both edges A and B are caulked, and the rivet heads also, if they leak, as at C. Fig. 153 shows how, in certain classes of boilers, the nipple or tube connections are caulked with a similar tool. A more satisfactory way of making the joints staunch and tight, known as **fullering**, which has largely superseded caulking, is shown in Fig. 155. The fullering tool, having a thickness at the end equal to that of the plate, is used in such a way that the greatest pressure due to the blows occurs at A, near the point, giving a clean finish, with less risk of damaging the plate. Fig. 154 shows a tool introduced by Mr. Webb, which combines features of caulking and fullering.

[1] The late Mr. J. Stansfield devised a multiple-punching machine, which he most successfully used in the construction of the floating docks designed by his firm.

[2] The great bridge which spans the Hooghly, made on the Thames some years ago, had the rivet holes for it, over a million in number, treated in this way.

[3] Too often this work is done by untrained youths, who are apt to bungle in overdoing a job that requires much care and not a little skill, if the efficiency of the joint is not to be impaired.

109a. The **sections of wrought iron and steel rolled bars**,[1] etc., used by the engineer are shown in Figs. 156 to 168, which should speak for themselves.

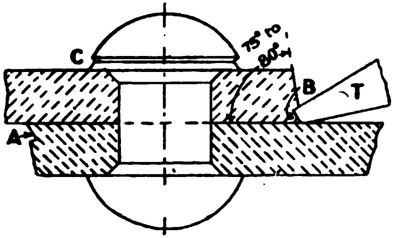


FIG. 152.—Ordinary caulking.

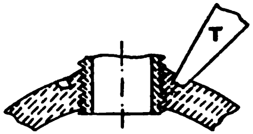


FIG. 153.—Caulking boiler connections.

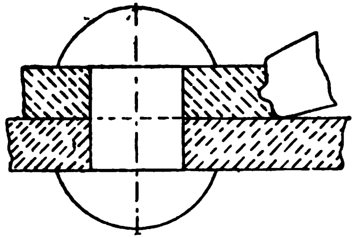


FIG. 154.—Combined caulking and fullering.

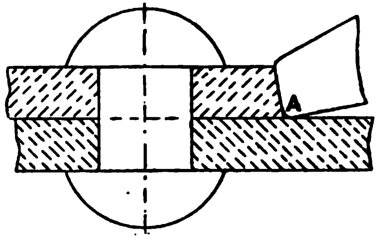


FIG. 155.—Fullering.

SECTIONS OF WROUGHT IRON AND STEEL ROLLED BARS.

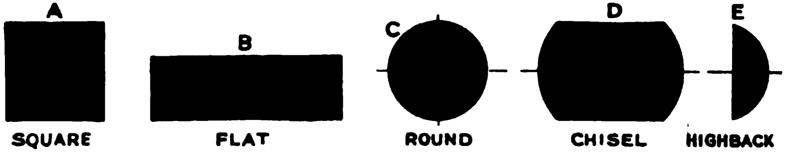


FIG. 156.—Sections of rolled bars.

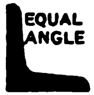


FIG. 157.

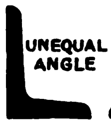


FIG. 158.

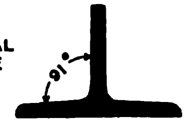


FIG. 159.—Tee.

FIG. 160.—Channel.

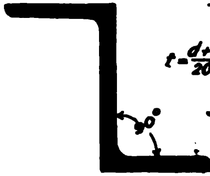


FIG. 161.—Zed.

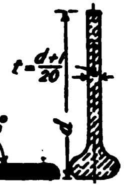


FIG. 162.—Bulb plate.

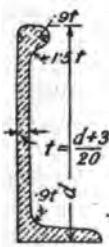


FIG. 163.—Bulb angle.

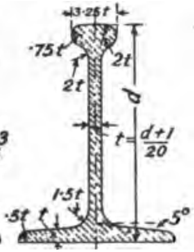


FIG. 164.—Bulb tee.

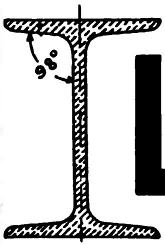
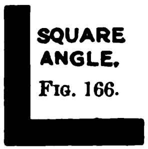


FIG. 165.—Beam.



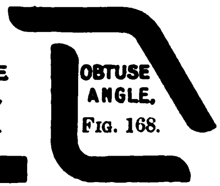


FIG. 167.—Round back angle.

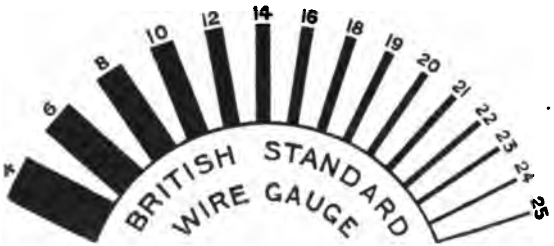
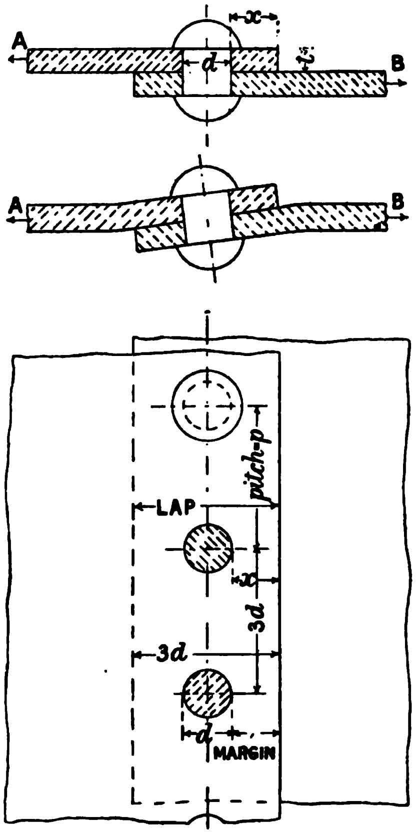


FIG. 169.—Showing the different thicknesses of iron and steel plates.

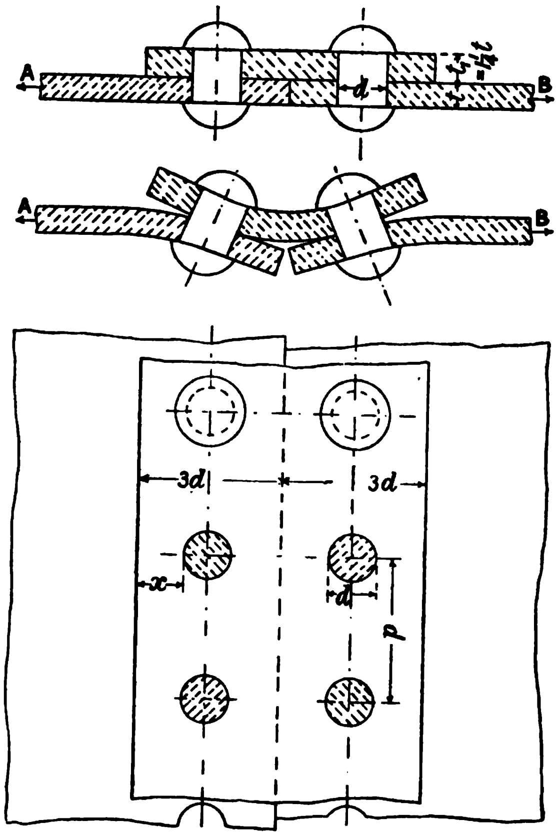
[1] For particulars of their limiting sizes, weights, etc., refer to the author's "Machine Design, etc.," p. 130.

109b. Comparative Cost of Bars and Plates.—The following give the approximate comparative cost of bars and plates:—

Flat, round, or square bars being	1·00
Angle and T-bars are	1·12
Plates are	1·18



Figs. 170 to 172.—Single riveted lap joint.



Figs. 173 to 175.—Butt joint. Single strap, single riveted.

110. Forms of Joints.—It will be convenient to briefly describe the principal joints before going into the question of strength, etc. The simplest form of riveted joints is the **lap joint**, with a single row of rivets, shown in Figs. 170 to 172. Although largely used, it has an obvious fault, for when the plates are in tension, owing to their not being in the same plane, a couple acts about the rivets, tending to bend the joint[1] to the form shown in Fig. 171 (and in Fig. 174 for butt joint with single strap); this being so, the plates are sometimes bent, before riveting, to approximately this form to reduce the bending action.

The **butt joint**, with a single *butt strap*, Figs. 173 and 175, has also this fault, and as it can only be caulked one side of the plate it should never be used for boiler purposes; indeed, it is very rarely used now.

Proportions.—The usual practice is to make the distance x between the side of a rivet and edge of the plate (called the **margin**) at least equal to the rivet diameter, thus making the minimum lap equal to $3d$, as shown, but in cases where the edges of the plates are more or less rough, $\frac{1}{4}''$ is added to this. We will go into the question of the pitch[2] p in another article.

[1] It is believed that *grooving* of boiler plates is sometimes indirectly due to this bending.
[2] The distance between the rivets, centre-to-centre, measured along the rivet lines.

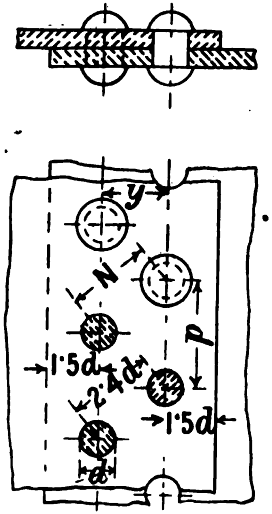
Figs. 176 and 177 show a double riveted (zigzag) *lap joint*. The **diagonal pitch** N should not be less than $2{\cdot}4d$, which is the German Lloyds' rule. According to Kennedy, in a double riveted butt joint, the net metal, measured zigzag, should be from 30 to 35 per cent. greater than that measured straight across, *i.e.* the diagonal pitch N should be $\frac{2}{3}p + \frac{d}{3}$. The Board of Trade rule for the distance y between the rows of rivets is $y = \frac{\sqrt{(11p + 4d)(p + 4d)}}{10}$. A rough rule may be, diagonal pitch $= \frac{p}{1{\cdot}3}$

110a. Chain Riveting.—When two or more rows of rivets are arranged with the rivets opposite one another, as in Figs. 178 and 179, we have chain riveting. In this case the distance y between pitch lines is usually at least $2d$, the **Board of Trade Rule** being $y = 2d$ to $2d + \frac{1}{2}''$.

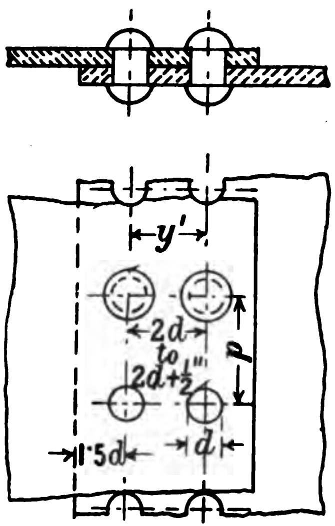
110b. The Combined Lap and Butt Joint.—This is shown in Figs. 180 and 181; it is an interesting joint that is used by some engineers for locomotive boilers. The figures speak for themselves. It will be noticed that the outer rows of rivets have twice the pitch of the middle row.

110c. Butt Joints with Double Straps.—Figs. 182 and 183 show a *double riveted* butt joint with *double butt straps*. It has been found by experiments that when the straps are made half the thickness of the plates (as it would appear they should be), the straps are then the weakest part.[1] This has led to the practice of making their thickness from $\frac{5}{8}t$ to t. The

DOUBLE RIVETED LAP.

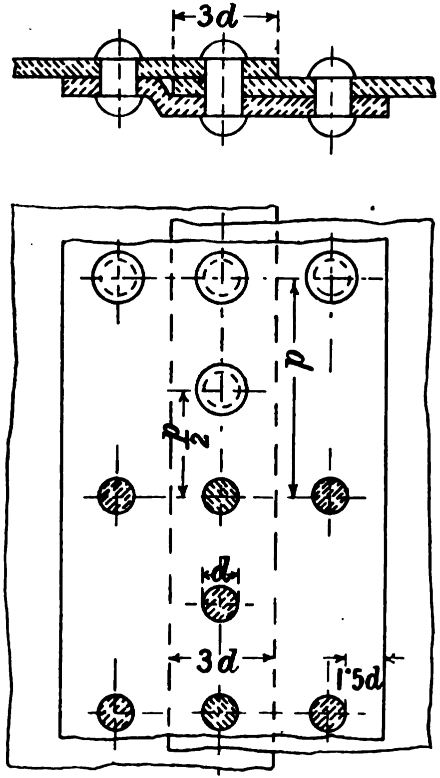


Figs. 176 and 177.—Zigzag.



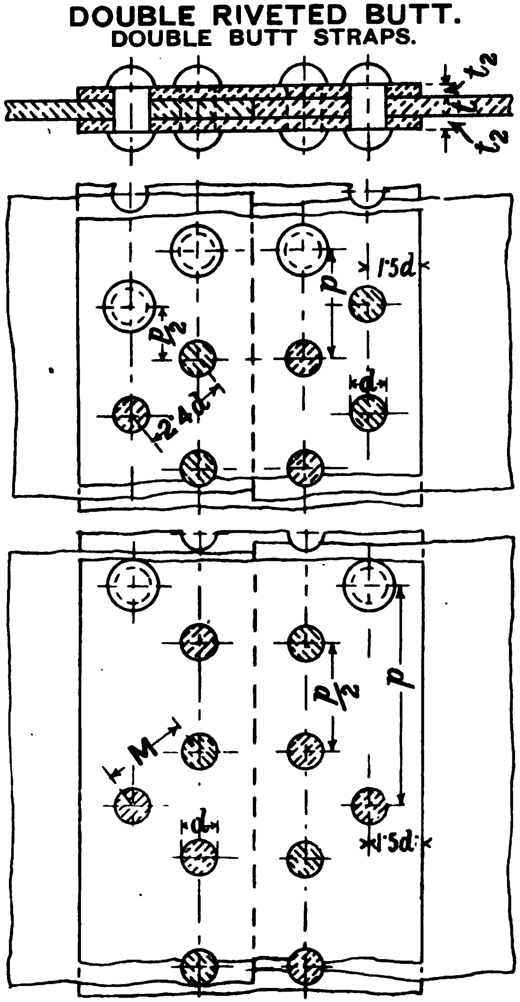
Figs. 178 and 179.—Chain.

COMBINED LAP AND BUTT.



Figs. 180 and 181.

[1] Doubtless this is due to an absence of absolute symmetry in the loading, and, therefore, to more than half the load coming on one of the straps.



Figs. 182 to 184.

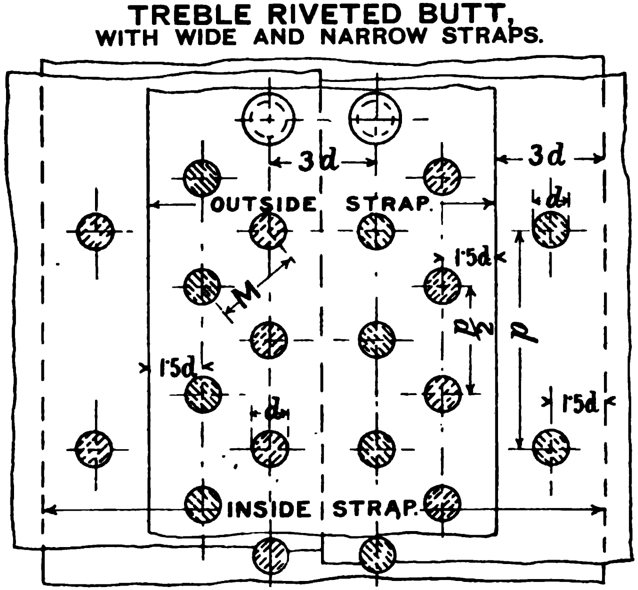


Fig. 185.

other proportions are shown on the figure. In Fig. 184 this joint is shown with the outer rows of rivets double the pitch of the inner ones, and we shall better see the advantage of this expedient when we come to deal with the question of strength.

Fig. 185 shows a form of joint on the same principle, with treble rivets and wide and narrow straps; it is used for large boilers working at high pressures. **A treble riveted butt joint** of a slightly different form is shown in Fig. 186, the edges of the **butt straps** being **scalloped** so that they may be efficiently caulked. It will be noticed that the **inner rows** of rivets have **half the pitch** of the outer rows, whilst in **quadruple riveted butt joints** the inner rows have one-third the pitch of the outer. And in all these joints the **diagonal pitch M** must be at least $2{\cdot}4d$; it is often more than this (touching almost $3d$ in some cases), which makes a safer joint.

There are many variations of the forms shown in the preceding figures, only the representative ones being selected for our purpose.

110d. Intersecting Riveted Joints.—Fig. 187 is a sketch of three rings of a cylindrical boiler,[1] the breadth of each ring being usually $3'$ to $3\frac{1}{2}'$, each ring being preferably made from one plate; the longitudinal joints arranged alternately to

[1] The most suitable material for boiler plates is *mild steel*, manufactured by the *Siemens-Martin open hearth process*, which is preferred to that produced by the Bessemer, because it is found to be more *homogeneous and reliable*.

the left and right of the top, which must be left clear for the fittings. When the rings are made with more than one plate the joints must clear the seatings of the boiler. At A the intersection of a longitudinal joint with a ring joint is shown, leading to a

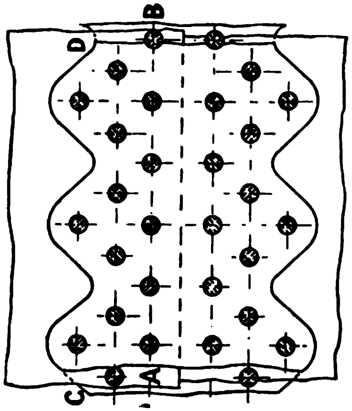


Fig. 186.—Treble riveted butt joints with scalloped edges.

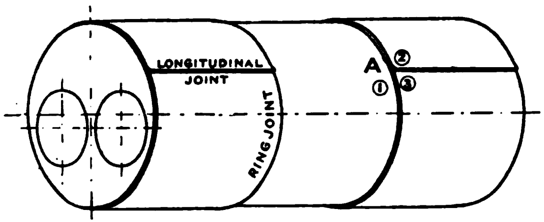
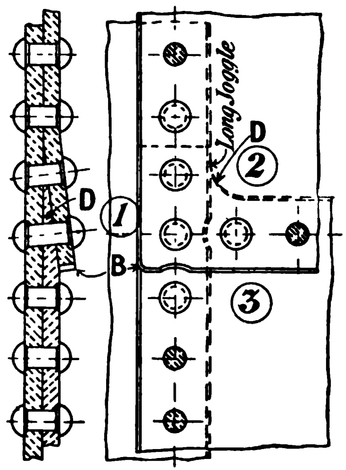
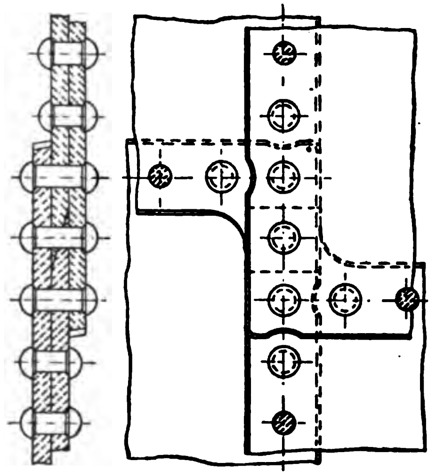


Fig. 187.—Junction of three plates in boiler shell.



Figs. 188 and 189.—Junction of three plates, single riveted lap.

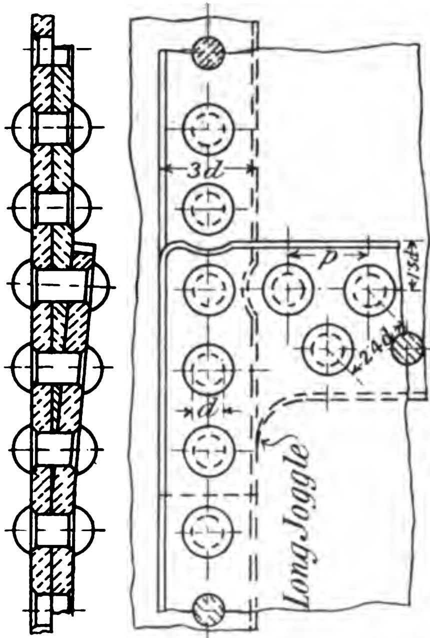


Figs. 190 and 191.—Junction of four plates, single riveted lap.

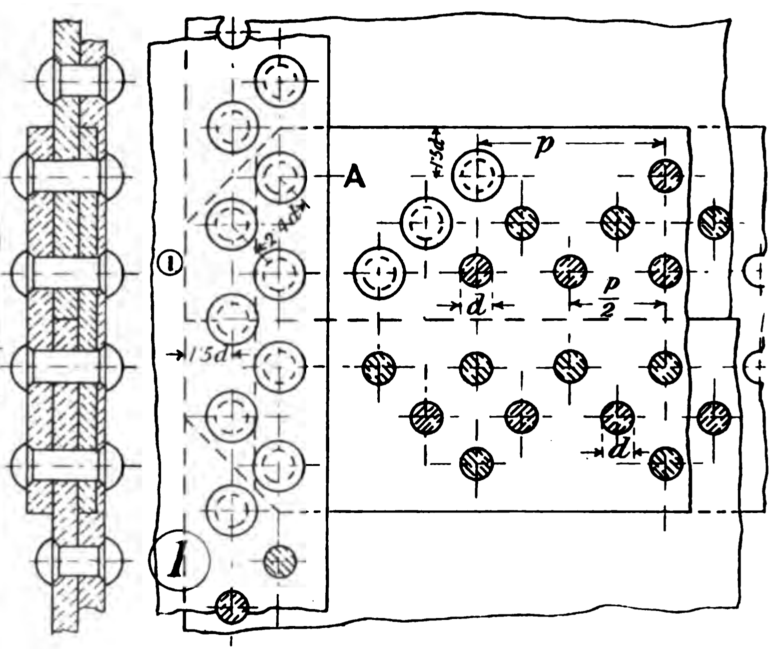
complication which has to be very carefully dealt with to ensure strength and staunchness, one necessitating the joggling or thinning by forging of the corner of one or more plates to form a solid joint throughout. Figs. 188 and 189 show how the junction is usually made when all the joints are *single riveted lap* ones, the corner B of plate 2 being forged and *set up*, and the corner of plate 3 hammered to a thin edge, forming a *long joggle*, and *tucked under* at D. But when two longitudinal joints are nearly in the same line, we get what is practically the **junction of four plates**, as shown in Figs. 190 and 191, which should now speak for themselves. The case which most often occurs in practice, certainly in boiler work, is the one where the longitudinal joints are made stronger than the ring joints, as they should be; we then in some cases get a single row of rivets in the ring joints, to double

rows in the longitudinal ones, as in Figs. 192 and 193, which are dealt with in the same way; or in more important work, where a higher efficiency in the strength of the joints is required, the ring joints are double riveted, and the longitudinal ones treble riveted, as in Figs. 194 and 195, where the strap A is planed down at the end wedge-shaped, to fit a recess machined out at the edge of plate No. 1.

JUNCTIONS OF THREE PLATES.



Figs. 192 and 193.—Single and double riveted lap.



Figs. 194 and 195.—Double riveted lap and treble riveted butt.

The principle is sometimes carried even a step further in very big work for great pressures, and we have double or treble rows for the ring joints, with quadruple ones for the others.

110e. Connections for Plates at Right Angles.—The simplest way of connecting two plates in this way is by an angle bar,[1] as shown in Figs. 196 and 197. The former is used to connect the front end of a Lancashire cylindrical boiler to the shell, and the latter was formerly the joint used for the back end of the shell. Usually, now, the back end is flanged and attached to the plate, as in Fig. 199. The mean thickness of the angle bar should not be less than the thickness of plate; usually it is a little over this. The holes should be as near to the adjacent *back* of the angle bar as it is practicable to place them,[2] but of course there must be room for the tools in riveting up. A good position would be X = $\frac{1}{2}$W (the width of the angle

[1] When the angle bar is in the form of a ring, it must be of very good quality to stand bending and welding.

[2] As far away from the thin edge as practicable. Usually, the front end is first riveted to the shell, the back end is afterwards offered on and the rivet holes marked. This enables any slight creep that may have occurred in building up the shell (and perhaps lengthening it $\frac{1}{16}$") to be dealt with without putting an initial strain on the end plate.

ring), but generally it has to be nearer the centre of y. Fig. 198 shows how, by flanging an end plate, the angle bar can be dispensed with, but only plates of very good quality can be thus flanged, particularly to a small radius. The radius of the inner surface

JUNCTIONS OF PLATES AT RIGHT ANGLES.

FRONT END OF BOILER AND SHELL.

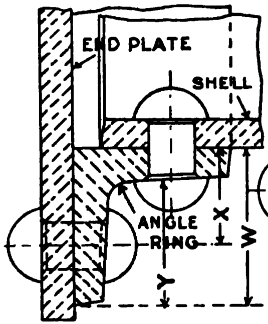


FIG. 196.

BACK END OF BOILER AND SHELL.

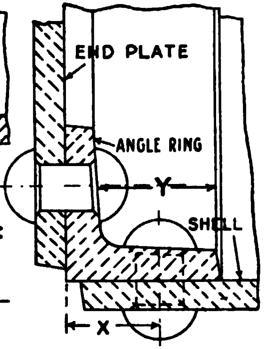


FIG. 197.

FLANGED END PLATE.

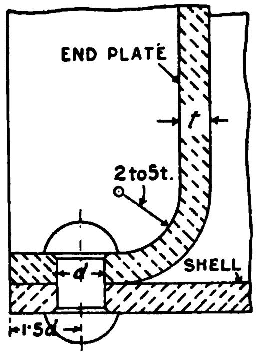


FIG. 198.

BOILER END, FLANGED END PLATE.

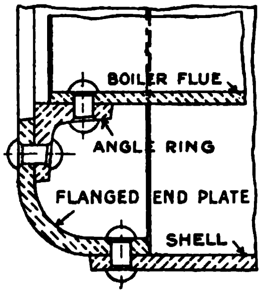


FIG. 199.

BOILER END, FLANGED END PLATE & FLUE.

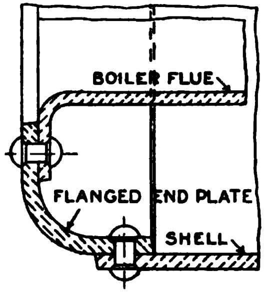


FIG. 200.

BOILER END, CORRUGATED FLUE.

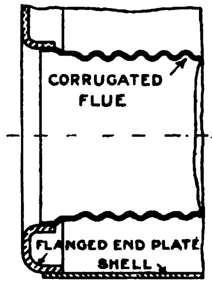


FIG. 201.

should not be less than $4t$, but in exceptional cases it is made as small as $2t$. Figs. 199 and 200 show how boiler flues are

FLUE CONNECTIONS.

TEE RING.

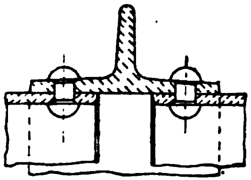


FIG. 202.

BOWLING RING.

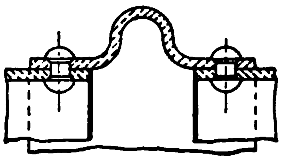


FIG. 203.

ADAMSON'S RING

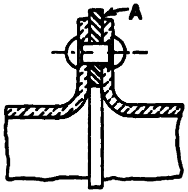


FIG. 204.

PAXMAN'S JOINT

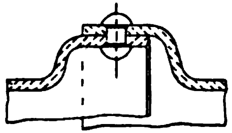


FIG. 205.

FLUE STIFFENING RINGS.

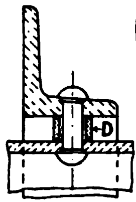


FIG. 206.

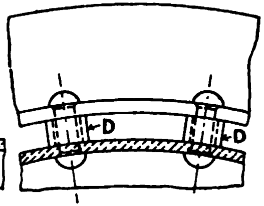


FIG. 207.

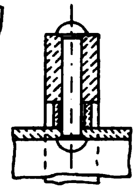


FIG. 208.

connected to end plates, the former the oldest, and probably the best, connection. When the end is flanged, as in Fig. 200, the

root of the flange is often too weak to bear the strains arising from the expansion and contraction, and sooner or later grooves and becomes fractured. An additional angle plate, giving two thicknesses at the flange, has been tried, but without success. So, on the whole, makers have found that the old rings, which are strong, and easily made and repaired, are the best. The connection of flue to the end plates is usually the same at both ends, and Fig. 201 shows the connection of a corrugated flue to a flanged end plate.

110f. Flue Connections.—Fairbairn's experiments show that the function of a flue joint should be to connect two sections or lengths of a flue in such a way as to give it longitudinal flexibility and circumferential rigidity, and Figs. 202 to 205 show some typical flue connections, but Fig. 202, the **Tee Ring**, is too rigid longitudinally for ordinary plain flues, and the rivet heads are exposed to the fire, as they are in the **Bowling Ring** or **Bolton Hoop**, Fig. 203, which has the advantage of being very flexible longitudinally.[1] **Adamson's joint**, Fig. 204, is a good one, being very rigid circumferentially, it has a proper amount of flexibility longitudinally; the ring plate A (about $\frac{3}{8}''$ to $\frac{1}{2}''$ thick) projecting a little beyond the flanges for caulking purposes. Fig. 200 shows how this flue is connected to the end plate. In the **Davey-Paxman joint**, Fig. 205, there are rivet heads inside the flue, but they are clear of the run of the burning gases. When a flue *section* is very long, it may be stiffened either by an angle ring, or solid ring; the former (shown in Figs. 206 and 207) is kept clear of the flue by distance pieces D, and riveted to the flue; the pitch of the rivets being about 7″. Fig. 208 shows the latter arrangement, with a solid ring instead of the angle ring. There are several variations of the above to be occasionally met with.

110g. Connecting Parallel Plates.—The lower part of the fire-boxes of vertical boilers, locomotives, and certain other boilers, are connected to the external shells as shown in Figs. 209 to 212. The simplest and most popular, particularly for locomotives, is 209; it is easily riveted and caulked. This joint is also used round the opening for the furnace door. In Fig. 210 a channel iron is used, but this requires the finest material and workmanship to forge the corners, and the rivets are not so get-at-able. The principal objection to the **Z**-bar in Fig. 211 is that the inner rivets cannot be caulked. Fig. 212 is much used for vertical boilers, but is unsatisfactory, as the sediment lodges in the recess S and causes corrosion.

FIRE-BOX CONNECTIONS.

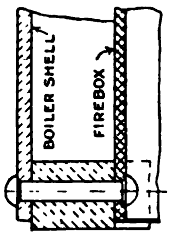


Fig. 209.

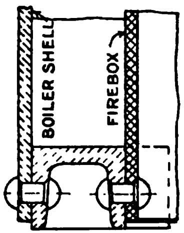


Fig. 210.

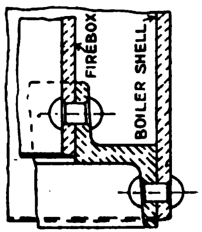


Fig. 211.

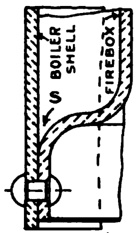


Fig. 212.

110h. Strength of Riveted Joints.—Let us first consider what may happen if we take a simple joint, such as the single riveted lap of Figs. 170 and 172, and assume that it has been tested till it fails. For this purpose we may deal with a strip representing a length of the joint equal to the pitch of the rivets, and it will be convenient in working examples to assume that we are dealing with steel plates, and rivets of a quality generally used for boilers. Other values from the Tables can be substituted as required. Then—

Let p = pitch of rivets.
„ S = strength of a strip of the joint of length p.

[1] Notwithstanding the exposure of the heads, some engineers, on the whole, prefer this joint to Adamson's.

Let d_n = diameter of rivet before riveting, or nominal diameter.
„ d = diameter of rivet after riveting.
„ f_t = tensile strength of material of plates per square inch = 28 tons for steel.
„ f_s = shearing strength of rivets per square inch = 23 tons for steel.[1]
„ f_c = crushing strength of plates and rivets at the hole = say, 46 tons per square inch.
„ t = thickness of plates.
„ η = efficiency of joint.

FAILURES OF RIVETED JOINTS.

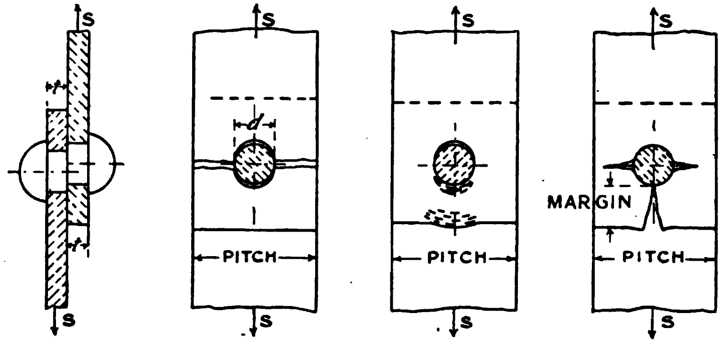


FIG. 213. FIG. 214. FIG. 215. FIG. 216.

The joints may fail—

(*a*) By **rivet shearing**, as in Fig. 213.

Then $S = d^2\frac{\pi}{4}f_s$ and $f_s = \frac{4S}{d^2\pi}$ as the rivet is in single shear.

(*b*) By the **tearing of the plates** between the rivets, as in Fig. 214.

Then $S = (p - d)tf_t$, and $f_t = \frac{S}{(p - d)t}$

(*c*) By the **plastic flow** of the material of the plate or rivet under compression, as in Fig. 215.

Then,[2] $S = dtf_c$, and $f_c = \frac{S}{dt}$

(*d*) By the **plate breaking in front of the rivet**, as in Fig. 216.

This rarely happens, as, when the distance of the edge of the plate from the rivet, called the margin, is equal to d, there is an abundance of strength.[3]

As a rule, when a joint is tested to destruction, it fails either as in (*a*) or (*b*), so, when these are equal in strength we have—

$$d^2\frac{\pi}{4}f_s = (p - d)tf_t$$

Whence, $$p = \frac{\pi d^2 f_s}{4tf_t} + d \text{ for single riveted lap joints} \quad (18)$$

[1] Allowed by Board of Trade. A *treble riveted lap-joint*, $d = 0{\cdot}95''$, $t = 0{\cdot}77''$, failed by rivets shearing, giving $f_s = 23{\cdot}9$ and $f_t = 28{\cdot}6$ (plate and rivets of steel), when tested by Kirkaldy.

[2] The area of the plate resisting the pressure of the rivet is the *projected* area of the rivet, namely dt.

[3] Considering the plate in front of the rivet as a short beam of length d, depth M, and breadth t, encastre at the ends, and equating the shearing strength of rivet to the bending strength, $M = \frac{1}{3}d$ nearly, where M is the margin.

110i. EXAMPLE.—What should be the pitch of a single riveted lap joint with steel plates and rivets, $d = \frac{3}{4}''$, and $t = \frac{3}{8}''$ for **equal strength in shearing and tearing?** Also find the efficiency of the joint.

By Eq. (18)
$$p = \frac{22 \times 9 \times 23}{28 \times 16 \times \frac{3}{8} \times 28} + \frac{3}{4} = 1{\cdot}72$$

Ans. $p = 1{\cdot}72''$ or $1\frac{3}{4}''$ bare. (In practice this would be made $1\frac{3}{4}''$.)

$$\textbf{Efficiency of joint} = \frac{\text{sec. area of plate at rivet}}{\text{sec. area of strip}} = \frac{(p-d)t}{pt} = \frac{p-d}{p} \quad \ldots \quad (18\text{A})$$

$$\therefore \eta = \frac{1{\cdot}72 - 0{\cdot}75}{1{\cdot}72} = 0{\cdot}564 \text{ or } 56{\cdot}4 \text{ per cent.}$$

111. Maximum Value of d in Relation to t. Crushing Action.—When joints fail by crushing, as in Fig. 215, the crushing stress has been found to be both high and irregular, but the usual practice of taking it to be about twice the shearing stress appears to be a safe one for ordinary purposes.

That is, $f_c = 46$ tons per square inch[1] for boiler steel.

In the case of boilers, the greatest stress on the bearing surface will occur during a hydraulic test, and this will seldom reach one-third the ultimate strength,[2] so, probably, joints as they are ordinarily made are not much injured by compression, unless the stress gets rather near the ultimate strength. Indeed, this seems to be borne out by the behaviour of joints that have been tested to destruction.

So, if the ratio $\frac{f_c}{f_s} = \frac{2}{1}$, as we have seen it may do, then, if the **shearing strength** is to equal the **resistance to crushing**—

$$d^2\frac{\pi}{4}f_s = dtf_c, \qquad \text{whence } d = \frac{4tf_c}{\pi f_s}$$

or (the maximum value of d in terms of t is) $d = 2{\cdot}54t$ . . . . . (19)

A larger rivet will crush, and we can infer that *the crushing effect need not be considered when in* **lap joints** *d is less than* $2{\cdot}54t$, as it usually is.

Now, if we had been considering a rivet in **double shear**, as those are in Fig. 183, then we should have to remember that the strength of a rivet in double shear is not quite twice that of one in single shear.[3] The Board of Trade consider $1{\cdot}75$ should only be taken; so, following this rule, the above equation becomes—

$$1{\cdot}75d^2\frac{\pi}{4}f_s = dtf_c, \qquad \text{whence } d = \frac{16tf_c}{7\pi f_s}$$

or (the maximum diameter d in terms of t for **double shear**) $d = 1{\cdot}455t$ . . . . . (20)

[1] In the case of boilers, with a factor of safety of $4\frac{1}{2}$, this amounts to the working value of f_c being 10 tons per sq. inch nearly, which in the opinion of some engineers should not be exceeded; indeed, 8 or 9 tons only is usually allowed in bridge work.

[2] The *testing pressure* used by the Boiler Insurance Companies ranges from $1\frac{1}{2}$ to 2 times the working pressure.

[3] This is probably due to absence of absolute symmetry about the line of force in the joint, and to the want of absolute uniformity in the quality of the materials.

A larger rivet will crush, and, therefore, on these lines *the crushing effect need not be considered when in* **butt joints** *d is less than* $1{\cdot}455t$.

112. Best Diameter of Rivet in Relation to Thickness of Plate.—It can be easily proved that if the d be less than t there will be danger of the punch crushing, so this consideration alone would limit the smallness of all *punched holes* in relation to the thickness of the plates, but it is a Board of Trade rule that d shall not be less than t, and if this condition be satisfied *the relation of d to t is to some extent arbitrary and may be varied considerably within certain limits*, experience dictating what is expedient in different thicknesses and classes of work, the relationship varying somewhat with the kind of work,[1] material, and joint. But it can be shown that the size of d is fixed when the thickness of plate and efficiency of joint are given. As a guide, where experience is not available, the following should be useful:—

According to Unwin	$d = 1{\cdot}2\sqrt{t}$ to $1{\cdot}4\sqrt{t}$	(21)
Box's Rule is	$d = (1\tfrac{1}{4}t) + \tfrac{3}{16}''$	(22)
Kennedy's Rule[2] for lap joints is	$d = 2\tfrac{1}{3}t$.	(23)

In **girder work**, when the rivets join several plates of total thickness T, the diameter of rivet d may $= \frac{T}{8} + \frac{5''}{8}$.

The diameter d is taken to the nearest $\frac{1}{32}''$. But the rivet must be small enough when red hot to freely enter the hole; so it is made from $0{\cdot}03''$ to $0{\cdot}06''$, (or $d_n = d - 0{\cdot}04''$ say) smaller than the hole.

TABLE 2.—Size of Rivets suggested by the National Boiler Insurance Co.

Thickness of plate.	Diameter of rivet. Finished size.	Thickness of plate.	Diameter of rivet. Finished size.
inch.	inch.	inch.	inch.
$\frac{3}{8}$	$\frac{3}{4}$	$\frac{11}{16}$	$\frac{7}{8}$
$\frac{7}{16}$	$\frac{11}{16}$	$\frac{3}{4}$	$\frac{7}{8}$
$\frac{1}{2}$	$\frac{13}{16}$	$\frac{13}{16}$	$\frac{15}{16}$
$\frac{9}{16}$	$\frac{13}{16}$	$\frac{7}{8}$	$\frac{15}{16}$
$\frac{5}{8}$	$\frac{7}{8}$	$\frac{15}{16}$	$\frac{15}{16}$

[1] In girder work $\frac{3}{4}''$ rivets are generally used for plates under $\frac{1}{2}''$, and $\frac{7}{8}''$ rivets for $\frac{1}{2}''$ to $\frac{3}{4}''$ plates; and whatever sized rivet is used, the pitch can be adjusted to obtain an equality of shearing and tearing resistances when required. But in boiler work the pitch is restricted by the pressure of the steam, as the joint must be staunch enough to be steam tight, and we are confronted with the anomaly, that as we increase the pressure we must reduce the diameter of the rivet.

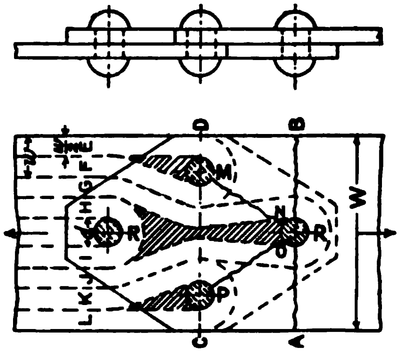
[2] Obviously, this size of rivet would be inconveniently large for thick plates and for treble riveting, but Kennedy rightly says that they (the diameters) should be made as large as possible.

113. Efficiencies of Riveted Joints.—In ordinary practice well-designed joints for boiler work should have the following efficiencies:—

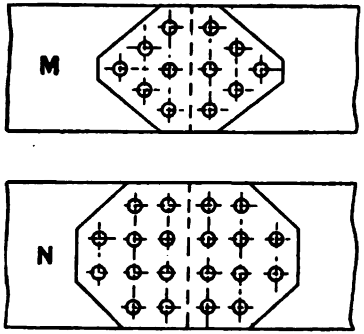
Single riveted η = 50 to 55 per cent.
Double riveted η = 65 to 70 per cent.
Treble riveted η = 80 to 85 per cent.

114. The Graphic Method of Designing Joints,[1] due to Schwedler, is in many cases very helpful; a simple example will suffice to explain the principle. Figs. 217 and 218 show a joint for a *tie bar;* such a joint would fail either by tearing across AB, or by shearing the four rivets. Then, for equal strength, a width of plate w, having a strength equal to that of one rivet, must be provided, as shown. To determine w

SCHWEDLER'S GRAPHIC METHOD.



Figs. 217, 218.



Figs. 219, 220.

we have $$wtf_t = d^2\frac{\pi}{4}f_s \text{ for } \textit{single shear}.$$

Whence $$w = \frac{d^2\pi f_s}{4tf_t} \quad \ldots\ldots\ldots \quad (24)$$

Then, if around each rivet a circle of $\frac{w+d}{2}$ radius be drawn, and flowing lines from these circles be drawn along the bar, it will be seen that each rivet has a portion of plate of equal strength allotted to it. And if S equals the strength of the solid bar, the strength of the joint will be $S - dtf_t$, the strength of the net section at AB. When the rivet lines CD and AB are close enough together, Kirkaldy found that the plate tears along DMNOPC, and that the further they are apart and the more easy flowing the lines of the strips w, the more likely is rupture to occur at AB. Obviously, the other things being the same, the smaller the rivet the greater will be the efficiency at this part; indeed, in some special cases the end rivets R have been made smaller than the others to increase the efficiency of the joint. Figs. 219 and 220 show at M and N two other joints (butt ones, with cover straps) of this type, also commonly used in bridge work.

EXERCISES.

Design, etc.

1. A single riveted lap joint, $\frac{1}{2}''$ plates, $\frac{7}{8}''$ rivets, both steel, f_s and f_t (the ultimate strength in shear and tension) being 23 and 28 tons per sq. inch respectively, find the most efficient pitch, also find the efficiency of the joint. *Ans.* $p = 1\frac{7}{8}''$, $\eta = 53$ per cent.

[1] For further information relating to the designing of riveted joints, refer to the author's "Machine Design, Construction and Drawing," p. 145.

2. Examine the joint described in exercise 7 below, and determine whether it would fail by the rivet shearing or the plate tearing, both being of steel.

3. A tie bar of rectangular section, $8'' \times \frac{3}{4}''$, is to be lengthened, a butt joint, with double straps, and nine rivets each side of the butt, arranged to give the strongest joint, being used. Sketch the joint, and determine the diameter of the rivets, the thickness of the straps and its efficiency, when f_s and $f_t = 23$ and 28 tons per sq. inch respectively.

4. 1″ diameter stays are to be used for the fire-box of a boiler; their working stress is not to exceed 4500 lbs. per sq. inch at their net section. The steam pressure being 140 lbs. per sq. inch, what distance apart should the stays be pitched?

Drawing Exercises.

5. Set out a double riveted lap joint, $\frac{1}{2}''$ plates, rivets $\frac{7}{8}''$, distance between rivet lines $1\frac{5}{8}''$, zigzag riveting. Scale full size.

6. Draw two views of a double riveted (zigzag) lap joint, $\frac{11}{16}''$ plates, $1\frac{1}{8}''$ rivets, distance between rivet lines $1\frac{7}{8}''$, pitch 3″. Scale full size.

7. Set out a treble riveted butt joint, double straps, one strap being double riveted only, as in Fig. 185. The distances of the three rivet lines each side of the butt (or centre of joint) are $1\frac{5}{16}''$, $2\frac{15}{16}''$ and $5\frac{5}{16}''$; plates $\frac{7}{16}''$, rivets $\frac{7}{8}''$, inner pitches $3\frac{1}{4}''$, outer pitches $6\frac{1}{2}''$. Scale full size.

Sketching Exercises.

8. Explain, with the assistance of sketches, in what respect the operation of *fullering* differs from *caulking*.

9. Make freehand sketches, in fairly good proportion, of the principal sections of bars used by the engineer.

10. Show by sketches how riveted *lap joints*, also *butt joints*, with *single butt straps*, become distorted when subjected to tensional strain. What defect in the form of the joint is the cause of this distortion?

11. Make a sketch of a combined lap and butt joint. What advantage has this joint over an ordinary lap one?

12. Make sketches showing how the ends of a cylindrical boiler are connected to the shell, both by angle rings and by flanging.

13. Show by freehand sketches three different forms of flue connections or joints, and point out their relative merits.

14. In how many different ways may a riveted joint fail? Illustrate your answer by sketches.

CHAPTER XII

BOLTS, NUTS, SCREWS, ETC.

115. It will now be convenient to give some attention to the *pair of elements* forming the fastening, which in the science of kinematics[1] is called a *screw-pair*, the simplest form of which is the common bolt and nut shown in Fig. 234. A fundamental feature of bolts and screws is that parts connected by them can be easily disconnected when required, and when it is realized what a great variety of work these interesting fastenings are used for, some idea can be formed of the multiplicity of forms and kinds that are in actual use; but for our purpose we shall give attention to a few of the most important only. Now, to completely specify some special form of bolt or screw it may be necessary to mention eight features, namely, (*a*) shape or form of the thread, (*b*) pitch or number of threads to the inch, (*c*) shape of head, (*d*) outline of body, barrel or stem, (*e*) size or diameter, (*f*) direction of threads (as *right-hand* or *left-hand*), (*g*) length, (*h*) material, as *iron*, *brass*, etc.

116. Forms of Screw Threads.—Figs. 221 to 233 show the best known threads used by the engineer. Fig. 221, a vee thread

FORMS OF SCREW THREADS.

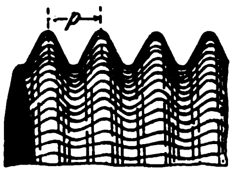


Fig. 221.—Whitworth's.

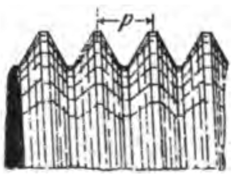


Fig. 222.—Seller's.

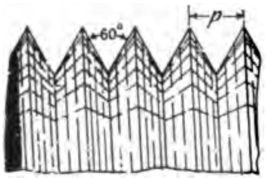


Fig. 223.—Vee.

Fig. 224.—Square.

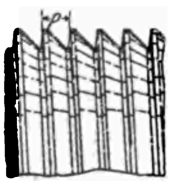


Fig. 225.—Buttress.

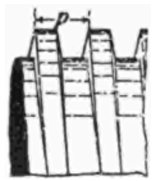


Fig. 226.—Acme standard.

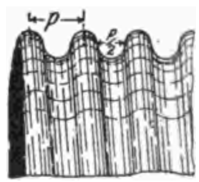


Fig. 227.—Round or knuckle.

slightly rounded at the top and bottom, is **Whitworth's**, *the standard British thread.* Fig. 222 is also a **vee**, with the top and bottom slightly flat; it is *Seller's*, and *the standard thread of America.* Fig. 223 shows a plain vee of angle 60°, used for most screws made of wood and for small brasswork. Fig. 224 is the **square thread**, Fig. 225 the **buttress thread**. Fig. 226 is the **Acme thread**, a square thread with a slight taper, to facilitate its being rapidly engaged and disengaged when used with a split nut, as in the screw-cutting lathe. In Fig. 227 is shown the *round top and bottom* or **knuckle thread**, largely used for railway

[1] Science of pure motion.

carriage couplings and hydrants, a very strong screw, not easily damaged when exposed to rough usage, but only suitable (for reasons that will be understood later) for special purposes.

117. Proportions of Threads.—It is not easy for the young engineer to imagine what a hopeless want of uniformity existed before the labours of the late Sir Joseph Whitworth[1] led to screw threads being standardized in this country (1857–61). The question of further standardizing screw thread and limit gauges engaged the attention of the Engineering Standards Committee in October, 1902, and a committee on Screw Threads and Limit Gauges was appointed. This committee issued an interim report in April, 1905, and their final report in 1907. A report on British Standard Nuts, Bolt-heads and Spanners was also issued in August, 1906, by the Engineering Standards Committee, and we shall refer to these reports where they affect current practice as we proceed.

In the old days it rarely happened that screws of the same nominal size made in different parts of the country had even the same pitch, therefore the all-important factor of interchangeability in the construction and repairing of engines and machines, which is now a fundamental feature of all good work, did not exist. Even now, when the screw threads proposed by Whitworth are universally used, screws varying in diameter, pitch, and number of threads from standard proportions, called *bastard* screws, are to be occasionally met with in old work. Fig. 228 shows the shape of our **Whitworth Vee thread.** The angle between the threads being 55°, and $\frac{1}{6}$ of the full depth of the triangle *abc* being rounded off at the top and bottom, to a radius of $0{\cdot}137329p$, as shown. But the full depth is 0·96 the pitch, so that the actual depth of the thread is $\frac{2}{3} \times 0{\cdot}96p = 0{\cdot}64p$, or, to be exact, $0{\cdot}640327p$. And if d = diameter of the screw at top of the threads, Fig. 230, and d_1 = diameter at bottom of the threads (the *net* diameter),

then **the diameter** $$d_1 = 0{\cdot}9d - 0{\cdot}05, \text{ nearly} \quad (25)$$

and if n = number of threads per inch,

Then[2] $$p = \frac{1}{n} = 0{\cdot}08d + 0{\cdot}04, \text{ nearly} \quad (26)$$

A series of **finer pitches**[3] (to supplement the Whitworth series) suitable **for bolts for connecting rod ends**, and piston-rod heads, is now in use, known as the **British Standard Fine Screw Threads** (B.S.A.). For screws up to and including 1″, the pitches being based upon the formula $p = \frac{\sqrt[3]{d^2}}{10}$, where d = full diameter of thread. And for sizes above 1″ and up to 6″, $p = \frac{\sqrt[8]{d^5}}{10}$.

Definitions.—The following definitions are due to the Engineering Standards Committee:—

Effective Diameter of a Screw.—The effective diameter of a screw having a single thread is the length of a line drawn through the axis and at right angles to it, measured between the points where the line cuts the slopes of the threads.

Core Diameter.—Twice the minimum radius of a screw, measured at right angles to the axis.

Full Diameter.—Twice the maximum radius of a screw, measured at right angles to the axis.

---

[1] Ramsden, in 1766, was one of the earliest to attempt to obtain extreme accuracy in originating screw threads in his dividing engine. The famous mechanician, Maudslay, subsequently took the matter in hand, and his labours did not cease until he had practically evolved the screw-cutting lathe.

[2] According to Briggs the relation of pitch and diameter of the Whitworth system is approximately—

$$p = 0{\cdot}1075d - 0{\cdot}0075d^2 + 0{\cdot}024$$

[3] Refer to author's "Machine Design, etc.," p. 193.

Crest.—The prominent part of the thread, whether of the male screw or of the female screw.
Root.—The bottom of the groove of the thread, whether of the male screw or of the female screw.
Slope of Thread.—The straight part of the thread which connects the crests and roots.
Angle of Thread.—The angle between the slopes, measured in the axial plane.

In Fig. 229 is shown **Seller's thread**, which we have explained is the standard shape adopted by America. The triangle being *equilateral*, the angle is therefore 60°, $\frac{1}{8}$ the full depth of the triangle being cut off top and bottom, as shown, to form

PROPORTIONS OF SCREW THREADS.

WHITWORTH'S THREAD

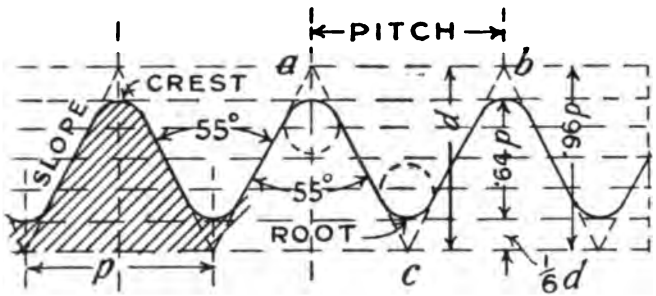


FIG. 228.

SELLER'S THREAD

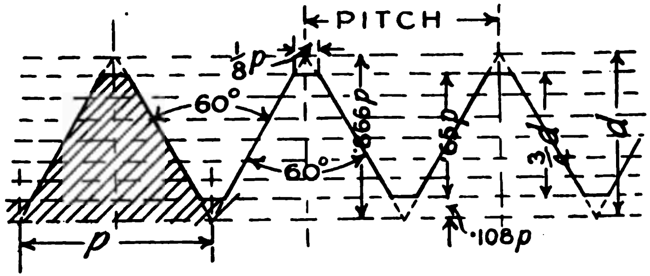


FIG. 229.

flats parallel with axis. So that the actual depth of the thread $d' = \frac{3}{4}d$, or $d' = \frac{3}{4} \times 0{\cdot}866p = 0{\cdot}65p$. The proportions of the **Square thread** (Fig. 224) are shown in Fig. 231, the pitch for standard screws being *twice* that for vee threads,

or, the **pitch for square threads** $$p = \frac{1}{n} = 0{\cdot}16d + 0{\cdot}08, \text{ nearly} \quad (27)$$

And, if d_1 = **diameter at bottom of threads**, as in the other cases,

Then $$d_1 = 0{\cdot}85d - 0{\cdot}075 \quad (28)$$

With this thread the thrust is very nearly parallel to the axis of the screw, and therefore there is no bursting strain on the nut, which is an important advantage. But the thread is more costly to produce than the vee thread, more particularly as it cannot be satisfactorily cut with dies. The figure shows the usual proportions of the thickness and depth of the threads.

Fig. 232 shows a modified form of the square thread known as the **Acme standard**, or 29° screw thread. It is used in machine tools where a disengaging nut is required, as previously explained.

The **depth of the thread** is (refer to Fig. 232) $$d' = \tfrac{1}{2}p + 0{\cdot}01 \quad (29)$$

And the width of the point of the tool for a screw or tap thread $= 0{\cdot}3707p - 0{\cdot}0052$

width of flat on top of the thread $= 0{\cdot}3707p$ . . . . (30)

This angle of 29° has also been generally adopted in cutting **worms** for gearing. Refer to Chapter on Spur Gearing.

The usual proportions of the **buttress thread** are shown on Fig. 233. This thread to a certain extent combines the important feature of the square thread already explained with the strength of the vee thread, but it has the disadvantage that it can only be efficiently used in one direction, namely, that which causes the thrust to act parallel to the axis, as shown by the arrows. In

PROPORTIONS OF SCREW THREADS.

WHITWORTH SCREW.

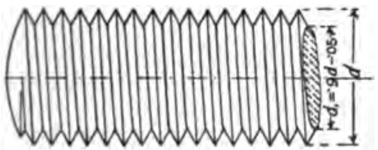


FIG. 230.

SQUARE THREAD

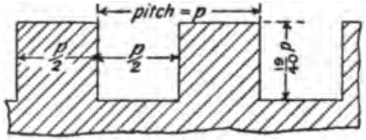


FIG. 231.

ACME SCREW THREAD

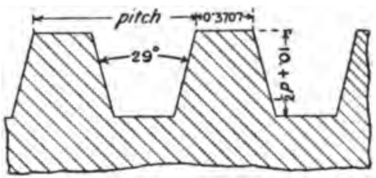


FIG. 232.

BUTTRESS THREAD.

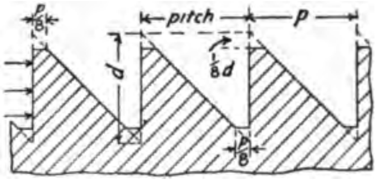


FIG. 233.

cases where there is little work to be done by it during a reversed motion, as in some presses, and when used on the breach blocks of large guns, the effect of the oblique thrust is negligible, and this is often the best form of thread for the purpose.

118. Drawing Exercise.—From a drawing point of view by far the most important detail is the bolt and nut, as any want of accuracy in presenting it mars the appearance of what otherwise might be a very good drawing, and offends the trained eye. Further, as the detail so often occurs on drawings, a real effort should be made to set it out in the usual *conventional way* shown in Figs. 234, 235, and 237.

Commence with the *Plan*, Fig. 237, by drawing the circumscribing circle (with a radius[1] equal to d, the diameter of the bolt) and the bolt circle (radius ½″), and from the latter draw projectors, cutting the former in a and b, join ab, and draw the chamfer circle, touching ab in c. The hexagon is then completed with the 60° set-square, making each of the other sides just touch the chamfer circle. Projectors from the corners e, f can now be drawn, and these, with projectors from a and b, give the indefinite elevation of the bolt body, and edges of nut and head. The thickness of the nut ($= d$) can now be set off, and with radius $1{\cdot}2d$, and centre on centre line, the arc fK can be drawn, and a line through these points gives M and N, which are used, as shown, to draw the arcs on the side faces;[2] the elevation of the nut is then completed by drawing the chamfers at 30°, to just touch the arcs. The head is drawn in the same way, making its thickness equal to $0{\cdot}9d$, whilst the point or end of the bolt is usually rounded with a radius $= d$. The screw threads are easily drawn in the conventional way shown, the slope being fixed by

[1] As we have explained, for drawing purposes (for 1″ bolts and under) it is convenient to make the diameter across the angles $= 2d$.

[2] A little practice will enable the student to draw these with considerable accuracy and facility, by feeling for the centre and radius, assuming tentative positions for the former till its true position is found.

PROPORTIONS OF HEXAGONAL BOLTS FOR DRAWING PURPOSES.

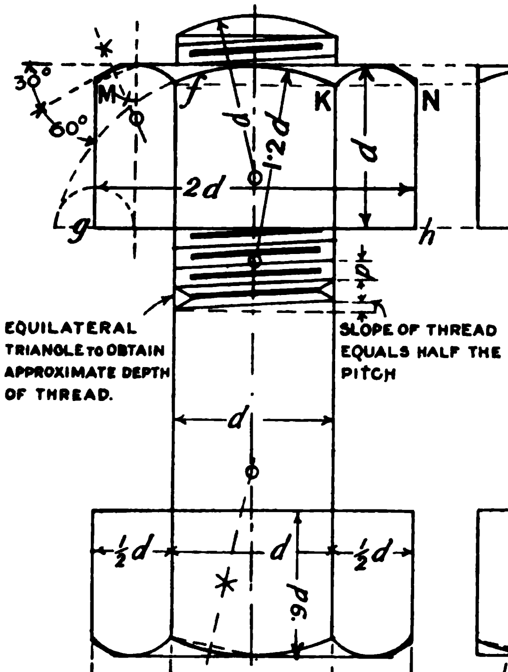


FIG. 234.

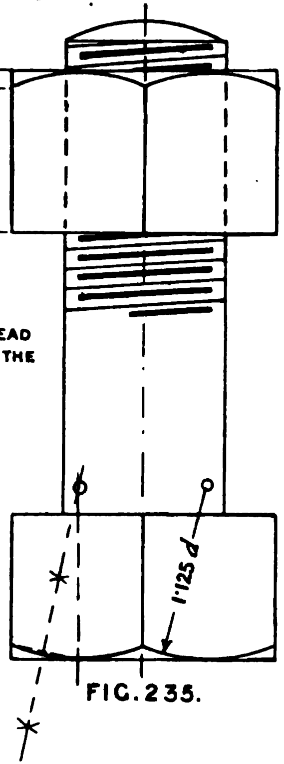


FIG. 235.

BOLT WITH SQUARE HEAD AND NUT.

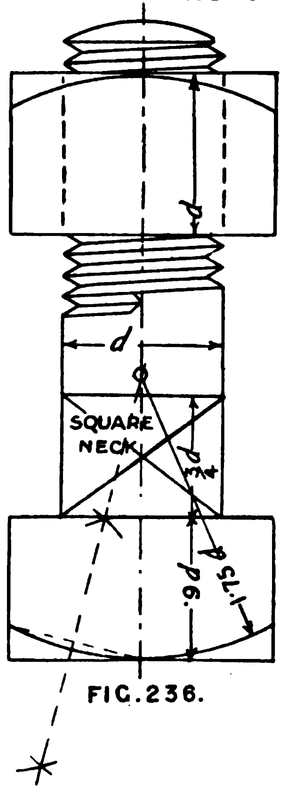


FIG. 236.

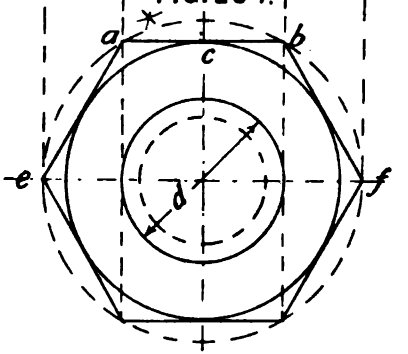


FIG. 237.

marking up ½ the pitch; the thick lines of course represent the bottom of the threads, and their diameter may be found by making the small equilateral triangle, of side equal to the pitch, which gives the approximate depth.

Fig. 236 shows a **bolt with square head and nut** and *square neck* to prevent the bolt rotating whilst screwing up; the proportions given in the table (No. 3), with the exception of the diameter across the angles, apply to these.

119. Various Types of Bolts, etc.—We may now give some attention to the various types of **bolts** and **bolt heads** in general use. Figs. 234, 235, and 237 show the form of the common hexagonal bolt and nut. The proportions of these are now **standardized**,[1] they are practically those given in Table 3, which are in common use. The practice of some manufacturers in the past has been to make **bright** nuts and heads, somewhat smaller in diameter than **black** ones, but this is very inconvenient, as, if for no other reason, it necessitates the use of two spanners for the same size bolt. However, as now standardized both the bright and black have the same *maximum* dimensions, the *minimum* dimensions fixed for the latter giving a larger *allowance*. For ordinary *drawing purposes* the

[1] Refer to Reports on British Standard Screw Threads, published by Crosby Lockwood & Son.

TABLE 3.—DIMENSIONS OF WHITWORTH'S 55° THREADS, HEXAGONAL BOLTS, NUTS, AND HEADS (BRIGHT).

Thickness of nut in each case = diameter of bolt.

Square nuts and bolts have the same proportions, with the exception of the diameter across the angles.

NOTE.—Refer to remarks in Art. 120 relating to this table.

Diameter of bolt of screw = d.		No. of threads per inch = n.	Diameter at bottom of threads = d.	Diameter across flats = D.	Diameter across angles = D × 1·155.	Thickness of bolt head = $\frac{7}{8}d = t$ (New Standard $t = 0·9d$) nearly.
$\frac{1}{16}$	0·0625	60	0·0411	0·212	0·2447	0·0547
$\frac{3}{32}$	0·09375	48	0·0670	0·280	0·3233	0·0820
$\frac{1}{8}$	0·125	40	0·0929	0·338	0·3902	0·1093
$\frac{3}{16}$	0·1875	24	0·1241	0·448	0·5173	0·1640
$\frac{1}{4}$	0·25	20	0·1859	0·525	0·6062	0·2187
$\frac{5}{16}$	0·3125	18	0·2413	0·6014	0·6944	0·2734
$\frac{3}{8}$	0·375	16	0·2949	0·7014	0·8191	0·3281
$\frac{7}{16}$	0·4375	14	0·3460	0·8204	0·9473	0·3828
$\frac{1}{2}$	0·5	12	0·3932	0·9200	1·0612	0·4375
$\frac{9}{16}$	0·5625	12	0·4557	1·011	1·1674	0·4921
$\frac{5}{8}$	0·625	11	0·5085	1·101	1·2713	0·5468
$\frac{11}{16}$	0·6875	11	0·5710	1·2011	1·3869	0·6015
$\frac{3}{4}$	0·75	10	0·6219	1·3012	1·5024	0·6562
$\frac{13}{16}$	0·8125	10	0·6944	1·39	1·6050	0·7109
$\frac{7}{8}$	0·875	9	0·7327	1·4788	1·7075	0·7656
$\frac{15}{16}$	0·9375	9	0·7952	1·5745	1·8180	0·8203
1	1·0	8	0·8399	1·6701	1·9284	0·875
$1\frac{1}{8}$	1·125	7	0·9420	1·8605	2·1483	0·9843
$1\frac{1}{4}$	1·25	7	1·0670	2·0483	2·3651	1·0937
$1\frac{3}{8}$	1·375	6	1·1615	2·2146	2·5571	1·2031
$1\frac{1}{2}$	1·5	6	1·2865	2·4134	2·7867	1·3125
$1\frac{5}{8}$	1·625	5	1·3688	2·5763	2·9748	1·4218
$1\frac{3}{4}$	1·75	5	1·4938	2·7578	3·1844	1·5312
$1\frac{7}{8}$	1·875	4·5	1·5904	3·0183	3·4852	1·6406
2	2·0	4·5	1·7154	3·1491	3·6362	1·75

NOTE.—The diameters increase by $\frac{1}{8}''$ from 1" up to 6", but the Engineering Standards Committee recommend that the following sizes should be dispensed with: $1\frac{5}{8}''$, $1\frac{7}{8}''$, $2\frac{1}{8}''$, $2\frac{3}{8}''$, $2\frac{5}{8}''$, $2\frac{7}{8}''$; in fact, all the odd $\frac{1}{8}$ths" up to 4" and, over 4", advances by $\frac{1}{2}''$.

The number of threads to the inch for all sizes remain unchanged. They are for the larger sizes: $2\frac{1}{8}''$ 4·5 threads to the inch, from $2\frac{1}{4}''$ to $2\frac{5}{8}''$ 4 to the 1", from $2\frac{3}{4}''$ to $3\frac{1}{8}''$ diameter $3\frac{1}{2}$ to the inch, from $3\frac{1}{4}''$ to $3\frac{5}{8}''$ diameter 3·25 threads to the inch, from $3\frac{3}{4}''$ to $4\frac{1}{8}''$ diameter 3 to the inch, $4\frac{1}{2}''$ diameter 2·875 threads, 5" diameter 2·75 threads, 5·5" diameter 2·625 threads, and 6" diameter 2·5 threads to the inch.

proportions shown in Figs. 234 and 235 may be generally used for the sake of convenience. For diameters up to 1″, heads and nuts may have the following approximate widths:—

$$\textbf{Approximate widths} \text{ for drawing purposes, up to 1'' diameter } \begin{Bmatrix} \text{Width across flats} = 1{\cdot}732d \\ \text{Width across corners} = 2d \end{Bmatrix} \quad \ldots\ldots\ldots\ldots \quad (31)$$

As **standardized,**[1] the maximum and minimum widths across the flats varies as follows:—

	Maximum.	Minimum.		Maximum.	Minimum.
For **bright nuts and heads**	$1{\cdot}5d + 1{\cdot}0''$	$1{\cdot}5d + 0{\cdot}970''$	for 6″ bolts to	$2d + 0{\cdot}025''$	$2d + 0{\cdot}020''$ for $\frac{1}{4}''$ bolts.
For **black nuts and bolts**	$1{\cdot}5d + 1{\cdot}0''$	$1{\cdot}5d + 0{\cdot}950''$	for 6″ bolts to	$2d + 0{\cdot}025''$	$2d + 0{\cdot}005''$ for $\frac{1}{4}''$ bolts.

120. Dimensions of Bolts and Nuts.—Table 3 gives the usual particulars of Whitworth's Bolts and Nuts up to a diameter of 2″. The diameters of the heads given are substantially the same as the recently standardized ones, but the latter are given with tolerances for limit gauging. The new standard gives the **heads a thickness** of approximately $0{\cdot}9d$. So in all cases where very exact dimensions are required, the standard tables should be referred to.

As we have seen, **finer threads**[2] than the Whitworth are now pretty generally used **for bolts of connecting-rod ends** and piston-rod heads; based on the formula, $pitch = \frac{\sqrt[3]{d^2}}{10}$ for sizes up to 1″, and $pitch = \frac{\sqrt[8]{d^5}}{10}$ for sizes 1″ to 6″.

121. Bolt and Screw Heads.—The following are typical examples of bolt and screw

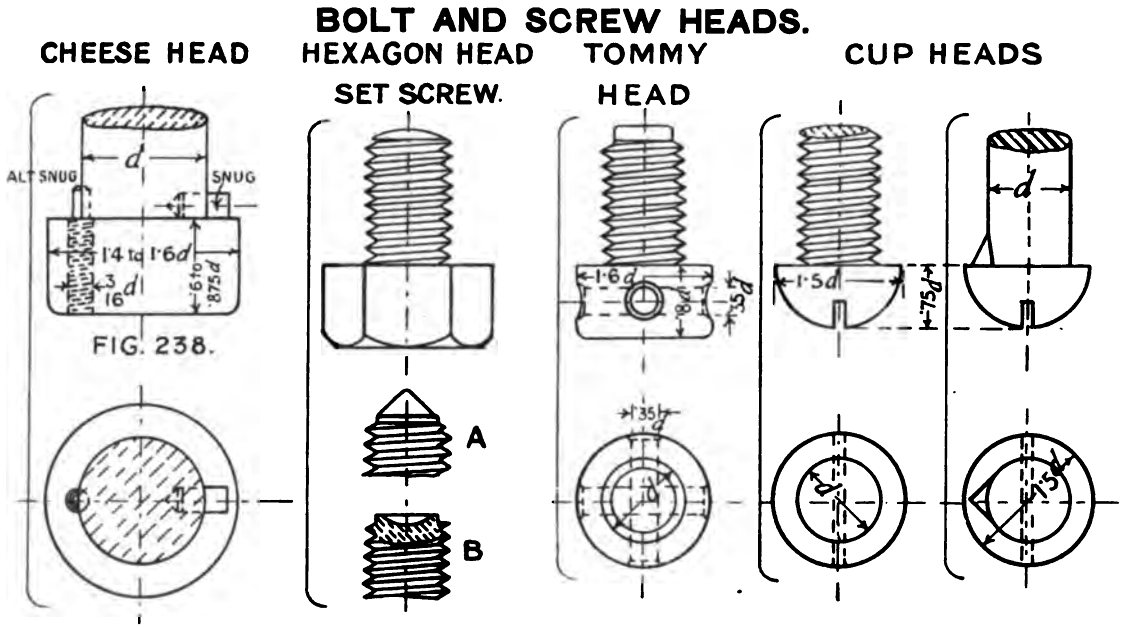


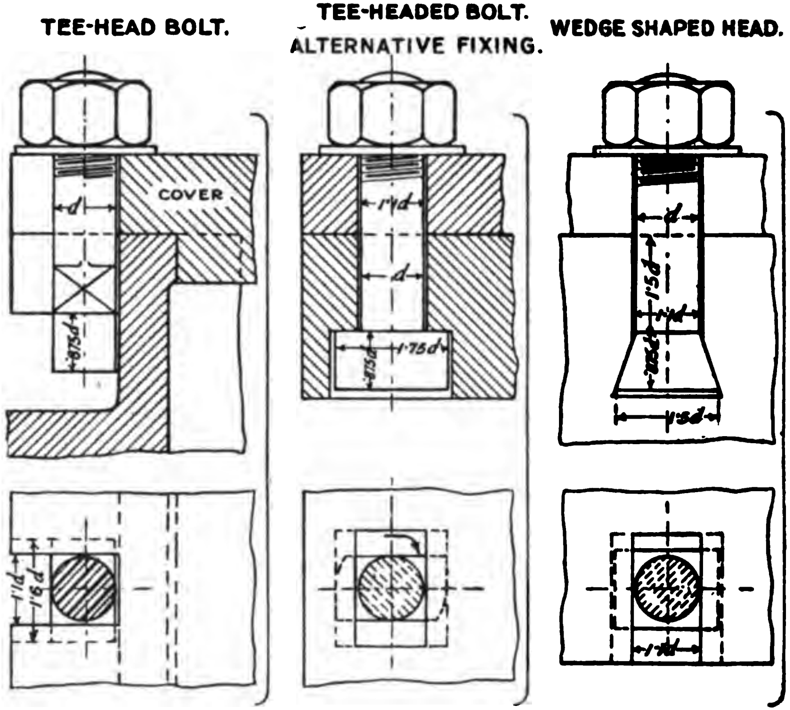
FIG. 238. FIG. 239. FIG. 240. FIG. 241. FIG. 242. FIG. 243.

[1] "British Standard Nuts, Boltheads," Published by Crosby Lockwood & Son, 2s. 6d. net.

[2] Refer to the author's "Machine Design, etc.," p. 193.

heads in general use. Figs. 238 and 239 show the **Cheese Head,** with two ways of fitting the *snug* which is required with round heads to prevent rotation whilst screwing up. Fig. 240 is an ordinary **Hexagon Head Set Screw,** the thread being run right down to the head. When this screw is used, to prevent any lateral movement, such as when it is used in a shaft's loose collar, the points are steel and either conical, as at A, or cupped out, as at B (refer to Art. 128). Fig. 241 is a **Tommy Head,** with cross holes, in which a round taper bar, called a tommy, is placed for screwing up. Figs. 242 and 243 are **Cup Heads**; such are used for **stove screws** and **coach bolts,** respectively. In the latter case the snug forged on prevents rotation. In Figs. 244 and 245 two views of an application of a **Tee-Head Bolt** are shown. These bolts are also largely used for holding down work on planing and other machines, the head fitting grooves in the table (as in the elevation, Fig. 246). The Figs. 246 and 247 also show an interesting way in which this bolt is sometimes used; the head is passed through the slot in the upper piece and then a quarter turn about the axis brings it into position so that the square corners prevent further rotation, as shown. In Figs. 248 and 249 are shown the **Wedge-Shaped Head.** This bolt is also used for holding down work on machines, etc. The **Hook Bolt,** in Figs. 250 and 251, is used in cases where there is not room for a bolt hole through one of the pieces to be connected, or in cases where a bolt hole would seriously weaken a piece; so they are used for attaching shaft hangers to the flanges of joists and girders. Figs. 252 and 253 show the head of an **Eye Bolt,** so arranged that when the piece to be fixed to the part A is fitted with an *open hole* the bolt can be rapidly swivelled into and out of position about the pin through its eye. This is a very useful arrangement, especially for pump work and for the valve covers of petrol motors, or for positions generally where the bolt, if loose, might fall out and get lost.[1] Figs. 254 and 255 show an arrangement of the **Eye Bolt** for swivelling in a plane parallel to the face of the work. Figs. 256 and 257 show the **Boss Head Bolt,** which must be used with judgment, as the wedge action of the head throws a bursting strain upon the adjacent metal. Fig. 258 shows a **Lifting Eye Bolt,** which is screwed into a hole near the centre of gravity of

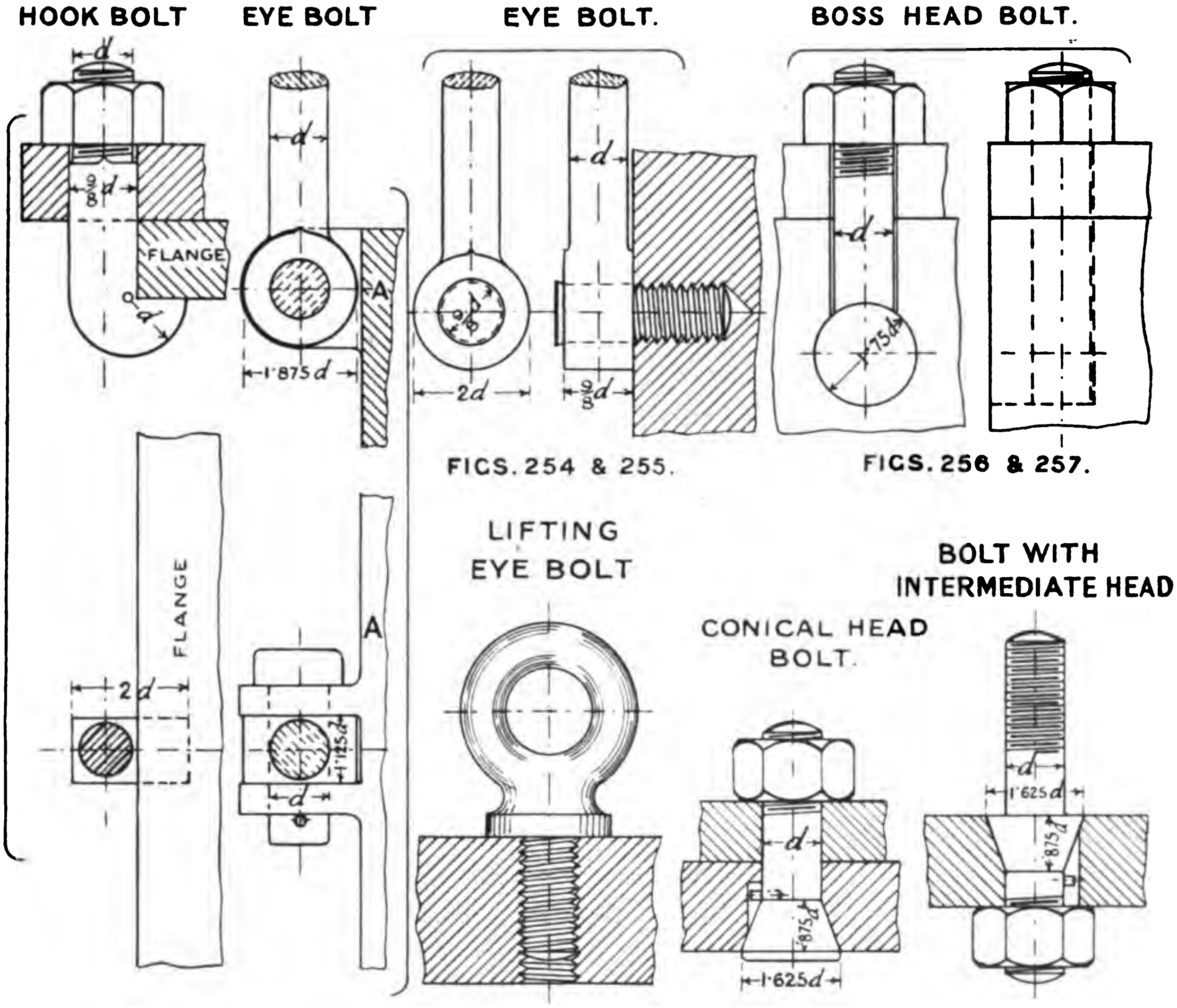
SPECIAL BOLT HEADS.



FIGS. 244, 245. FIGS. 246, 247. FIGS. 248, 249.

[1] The lugs on the casting A have, by an oversight, been made too thin.

SPECIAL BOLT HEADS, ETC.



FIGS. 250, 251. FIGS. 252, 253. FIG. 258. FIG. 259. FIG. 260.

cylinder covers, jackets, etc., for the attachment of a rope or chain for lifting purposes. Fig. 259 shows a **Conical-Headed Bolt,** with snug to prevent rotation whilst screwing up; this is used when there is no room for an ordinary head. In Fig. 260 we have a **Bolt with Intermediate Head** or flange; it remains in position when the top nut is taken off; in fact, it is a **combined Bolt and Stud.** Fig. 261 shows an **Ordinary Stud**; it is so fitted that it is a snug fit when screwed home in the flange F, the thickness t of which should not in any case be less than $1\frac{1}{4}d$, but $1\frac{1}{2}d$ is a better proportion. Studs are occasionally made with a round or *Square Collar*, the latter (shown in Fig. 262) can be used with a spanner for screwing up; it also forms a shoulder to screw home to on the flange. Fig. 263 shows a tapped hole for a **Forcing or Lifting Screw** C, closed by Set-Screw D, when out of use. Cylinder covers, junk-rings, valve-chest covers, etc., are fitted with these to break the joints prior to lifting with Eye Bolts. In Fig. 264 is shown an **Adjusting Screw** with a locking nut. It is used for a variety of purposes, notably for adjusting the position of sliding strips for the sliders of machine tools and lathes. When the part in contact with the end of the screw is arranged so that a movement at right angles to the screw can be made, then a **saddle-piece**[1] S, Fig. 265, is used to prevent a burr being formed which would prevent movement. The hole for the saddle-piece is made by an *arbor* tool worked through the screw hole. Fig. 266 shows a case where the bolts would have to be double-nutted, whilst in Fig. 267 is shown a case where, if A is a stud, it would not be practicable to use a *stud* at B, as the angle plate C could not be put on or taken off without using a *screw* B which can be withdrawn. Screws and studs screwed into tight-fitting holes, which do not penetrate the lower surface of the flange or piece, must have a groove running lengthwise across the threads to allow the air to escape from between the bottom of the hole and the end of the screw whilst the stud or screw is being screwed home.

USE OF STUDS.

ORDINARY STUD

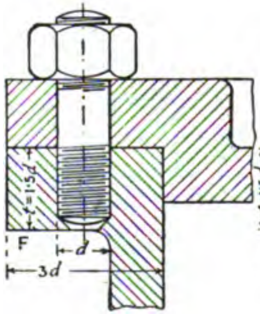


Fig. 261.

STUD WITH SQUARE COLLAR

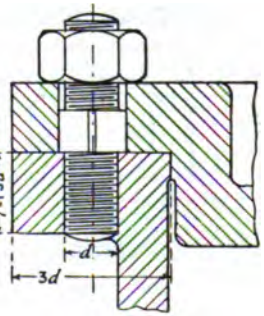


Fig. 262.

FORCING SCREW

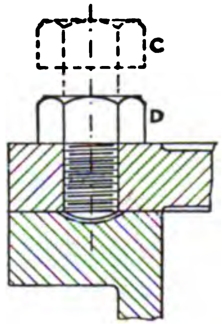


Fig. 263.

ADJUSTING SCREW.

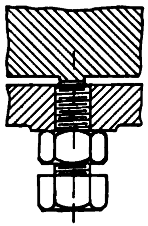


Fig. 264.

SET SCREW WITH SADDLE.

Fig. 265.

DOUBLE NUTTED BOLT

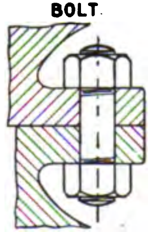


Fig. 266.

STUD AND SCREW.

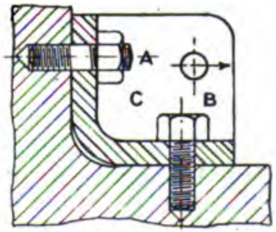


Fig. 267.

122. Special Nuts.—There are a great number of special nuts in use, a few of which we will call attention to. Fig. 268 is the **spherical seated** one, used to allow a small movement from normal position of the part on which it rests; it is much used on tool holders of lathes and machines. The **Flanged Nut**, Fig. 269, gives a larger bearing surface; it is suitable for use when the hole is sensibly larger than the bolt, but it is costly

[1] This is conveniently used to hold tight the bolt which holds the back centre in the loose headstock of some small lathes.

to make. In Fig. 270 is shown a **Flanged Cap Nut**, used to prevent fluid leakage past the screw threads. To prevent leakage at the bearing surface a *copper washer* may be used, as in Fig. 271. The ordinary **Thumb Nut** is shown in Fig. 272, and another form in Fig. 273.

123. Extra Thick Nuts.—When a nut has to be very often removed, as for instance when used on tool holders, it is usually made of steel, and at least one and a quarter times the ordinary thickness, to reduce the wear both of the threads and faces. It is also made thicker when of a softer and weaker material than the bolt (for instance, a gun-metal[1] or brass nut for an iron bolt, or an iron nut for a steel bolt), so that there may be as little wear of the bolt as possible. Nuts which have to be constantly taken off are *case hardened* to prevent the sides being unduly worn. But iron and steel studs should not be screwed into bronze, as they rapidly rust, especially when exposed to the action of sea water.

SPECIAL NUTS.

SPHERICAL SEATED NUT.

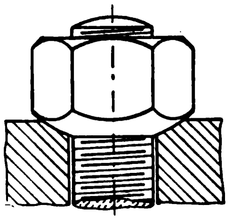


Fig. 268.

FLANGED NUT.

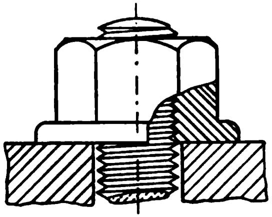


Fig. 269.

FLANGED CAP NUT.

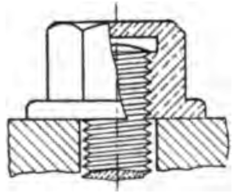


Fig. 270.

CAP NUT WITH COPPER WASHER.

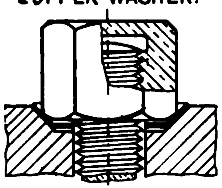


Fig. 271.

THUMB NUT.

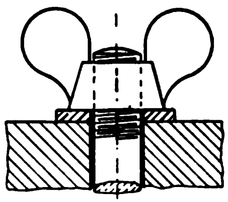


Fig. 272.

THUMB NUT

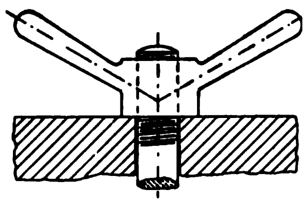


Fig. 273.

124. Locking Nuts and Arrangements.—No matter how perfect the fit of a nut on its bolt may be, when it is subjected to vibration, or to the jarring tremulous motion of machinery, the nut gradually works loose or tends to do so, and may, if there is nothing to stop it, work off the bolt. Now, one of the best known expedients to prevent this, and the one usually employed when pieces subject to rapid motion are connected by bolts, is the **Lock Nut**,[2] which is an extra nut screwed tightly down on to the ordinary one, as in Fig. 274, to jamb or *lock it* on the bolt in such a way that it will not work loose and gradually screw off the bolt. This lock nut is sometimes made half the ordinary thickness of a nut, on the assumption that it is only to jamb the other nut and take only a small part (if any) of the load, but a little consideration will satisfy the student that it is the *top nut which practically takes the whole load*, and of course the *thick nut* should be there (as in Fig. 274), as the *true lock nut*,[3] but spanners are rarely thin enough to take the lock nut when it is so thin and is placed at the bottom, and this has led to the growth of the faulty practice shown in Fig. 275. An obvious way out of the difficulty would be to make both nuts the full thickness, but there is not always room for this, and when there is it offends the eye, so the compromise of keeping the total thickness the same and making them both the same thickness, namely $\frac{2}{3}$ to $\frac{3}{4}d$, is one that is often met with. However, the standard arrangement is now the one shown in Fig. 274, and this

[1] For a gun-metal nut on an iron bolt that is fully strained the thickness of the nut should be $1{\cdot}9d$.
[2] Nothing completely satisfactory has yet been evolved to *lock* the nuts of rail fish-plate bolts, and prevent them being loosened by the shocks of passing trains.
[3] It is the practice of some engineers to arrange the nuts in this way, and to make the thickness of the bottom one equal to d and the top one equal to $\frac{7}{8}d$.

should always be used when convenient. Fig. 276 also shows the end of the bolt turned down to allow the nut to be screwed on and off easily,[1] and to more conveniently allow of a **split pin** to be used, where the bolt is subject to much vibration, to prevent the nuts working off. There are many.

125. Other Locking Arrangements.—The one shown in Fig. 277, the **Penn** or **Ring Nut,** is largely used, particularly for the bolts of connecting rod ends and the studs of piston rod heads; a circular recess being made in the cap A to receive the lower part of the nut, which is turned to suit, and grooved to allow the point of a set-screw to press on it and *lock it.* Fig. 278 shows an obvious variation, the collar being used to save recessing the cap, a pin P prevents rotation of the collar. One of the simplest ways of locking a nut is shown in Fig. 279, but to make a job of this the steel set-screw should have a hardened cupped point and be placed opposite a thread, then, if the fit of the set-screw be snug and it is screwed up with judgment, the nut is held tight and very little burr is raised on the screw.

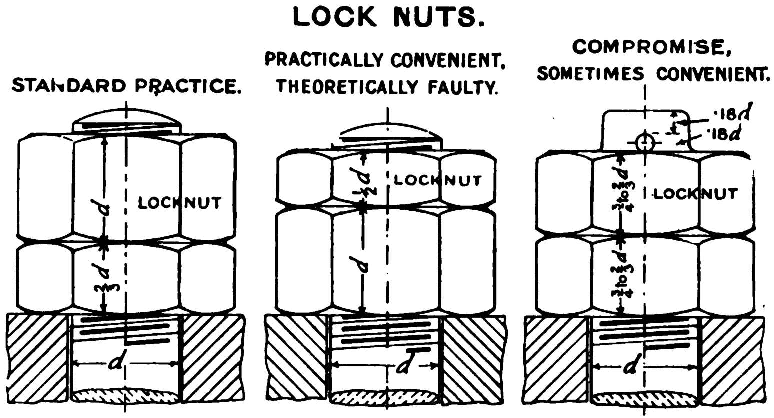


FIG. 274. FIG. 275. FIG. 276.

126. Wile's Lock Nut. The Use of Taper and Split Pins.—Figs. 282 and 283 show Wile's lock nut; it is sawn halfway through, and a set-screw at S draws the parts together after the nut has been screwed home, gripping the threads. For sizes smaller than 1″ the set-screw is not generally used, a hammer blow being sufficient to slightly close the saw cut before the nut is screwed on. **Locking by Pins**; the common expedient of putting a split pin through the bolt just above the nut, as in Figs. 284 and 285, prevents the nut screwing off, but hardly locks it. On the other hand, a **taper split pin** (taper about $\frac{1}{4}$″ to the foot) fitted to a reamered hole through the nut and bolt as in Figs. 286 and 287, with the split end opened out, forms an *absolute lock,* and piston ends are often fitted this way. The **standard proportions** of ordinary taper pins are given in Table 1.

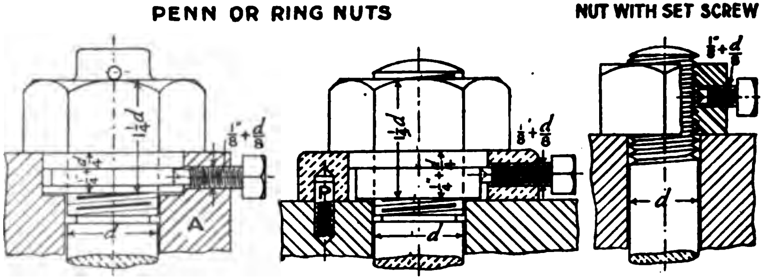


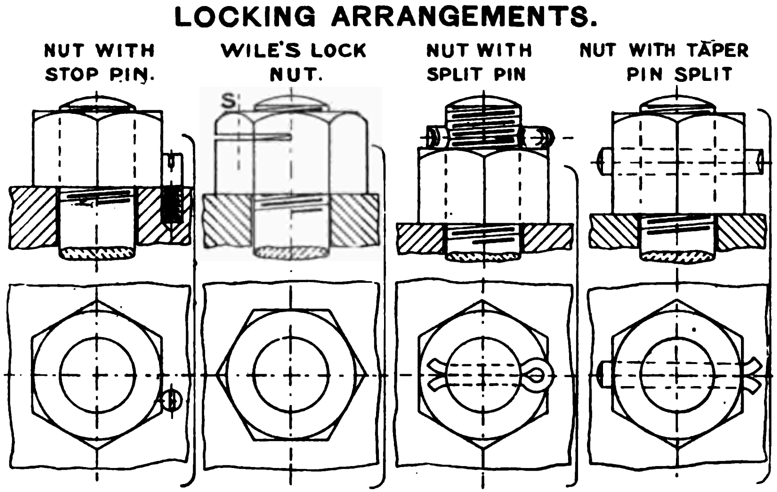
FIG. 277. FIG. 278. FIG. 279.

126a. The **Capstan Nut** or **Castle Nut** (Fig. 287A) is largely used for locking purposes in motor-car work, and on jobs generally that are subjected to sudden shocks and much vibration. It consists of an hexagonal nut with a portion turned off making a circular collar, through which rectangular slots are made, and into which, after the nut has been adjusted, a round or rectangular

[1] This arrangement is especially necessary in large horizontal screws, such as the screwed end of a propeller shaft.

cotter with split ends is fitted through nut and bolt. The standard proportions recommended are, D = width across flats $- \frac{1}{16}''$, $T = 1{\cdot}25d$, $H = 0{\cdot}75d$, and $t = 0{\cdot}4375d$, the radius R may $= \frac{d}{8}$, and $W = 0{\cdot}25d$.

127. Foundation Bolts.[1]—Figs. 288, 289, and 290 show three different arrangement of heads for fixing into stone work. The taper head in Fig. 288 is jagged, and molten lead or sulphur is poured into the taper hole to fill the space between head and stone. Where great strength is required four parallel bars, or *keys*, are used in addition, as in Fig. 289. For temporary fixing, or for lifting heavy blocks of stone, the **Lewis Bolt** (or taper bolt) head is used. It has a single movable key (Fig. 290) made with a slight taper and a head H to facilitate withdrawal. The total **taper** in each bolt is **1½″ to the foot.**



Figs. 280, 281. Figs. 282, 283. Figs. 284, 285. Figs. 286, 287.

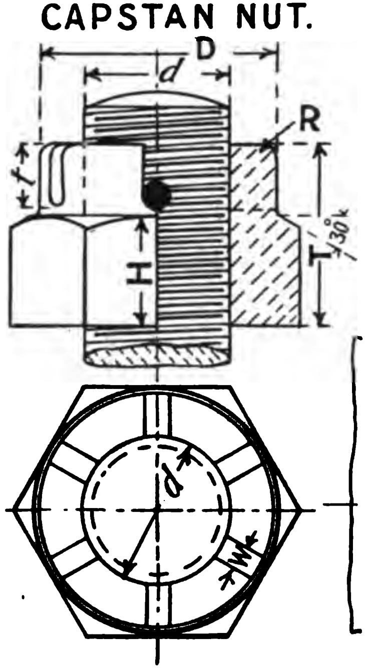


Fig. 287a.

128. Loose Collars.—Whenever end movement of a shaft or spindle is to be prevented collars are used. These may be part of the shaft itself, or **loose,** as in Figs. 291 and 292,[2] in which case they are secured to the shaft (in contact with the bearing at F), and their end movement is prevented by set-screws. Ordinary loose collars are made without the *boss* shown; they can then be easily turned all over.

129. Set-Screws.—The points of the set-screws are usually of hardened steel, or if of wrought iron they are case-hardened. The shape of the point may be either *conical,* as shown at C, Fig. 291, or rounded, as at A, or cupped, as at B. The rounded point does

[1] Also refer to Figs. 372, 373, and 374.
[2] Also refer to Fig. 86.

the least amount of damage to the shaft, and it has a good holding power.[1] When the conical point C is used for cases where there is considerable holding power required, a conical hole is usually drilled in the shaft to receive it. Set-screws are also used in some cases where the *rotation* of a piece on a shaft is to be prevented.

130. Washers.—When the seating of a nut is rough or uneven a washer is used to provide a smooth surface for the nut to turn

FOUNDATION BOLTS.

RAG BOLT.

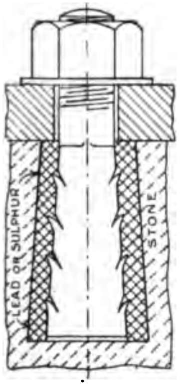


FIG. 288.

RAG BOLT WITH KEYS

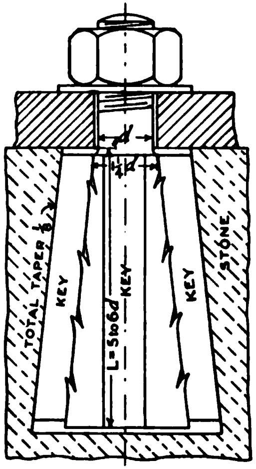


FIG. 289.

LEWIS BOLT

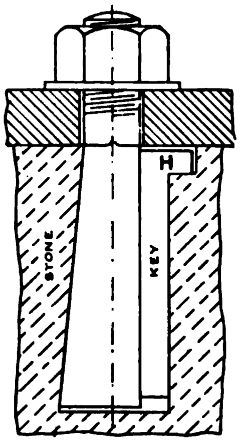


FIG. 290.

LOOSE COLLAR.

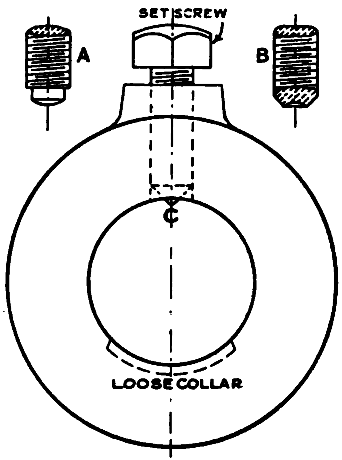


FIG. 291.

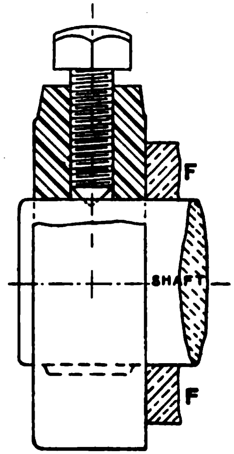


FIG. 292.

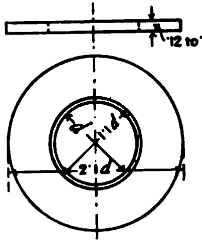
on. Washers are also used to spread the pressure of the nut over a larger seating when the material is weak enough to require it. Figs. 293 and 294 show an ordinary rough black washer (plain or square edge) punched out of sheet iron. When washers are used with bright nuts they are turned and usually finished with a slight *chamfer*, as in Fig. 295, A being *bevelled*, B rounded, and C hollowed. The coiled spring washer, Fig. 296, forms a *lock* for the nut when the latter is screwed down on it. Unscrewing is more or less prevented by the sharp edges N and M tending to cut into the nut and bearing surface respectively, and by the washer taking up any slight back-lash, and in so doing reducing the tendency to work off due to vibration. Obviously this washer should not be used when the nut is to be frequently removed. The **proportions** shown on Figs. 293 and 294 are about the ordinary ones, but for wood and other soft materials they must be made proportionately larger in diameter, and thicker.

[1] Professor Lanza, in experimenting on the holding power of set-screws, found that this form offered the greatest resistance to sliding; a ⅝" screw, end rounded to ½" radius, having a mean *holding power* of 2912 lbs., the mean holding power of the *cupped* screw B being 2470 lbs.

131. Screws with Multiple Threads.—The *pitch* of a screw may be defined as *the distance its nut advances for one revolution.* Now, we have seen (Eq. 27, Art. 117) that with a single threaded screw of fixed diameter d, Fig. 297, the greater the pitch the

WASHERS.

BLACK WASHER



FIGS. 293, 294.

BRIGHT WASHERS

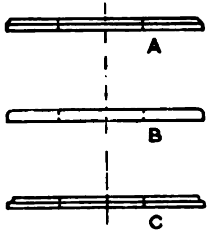


FIG. 295.

GROVER'S COILED SPRING WASHER

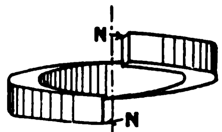


FIG. 296.

SCREWS WITH MULTIPLE THREADS.

SINGLE THREAD DOUBLE THREAD TREBLE THREAD

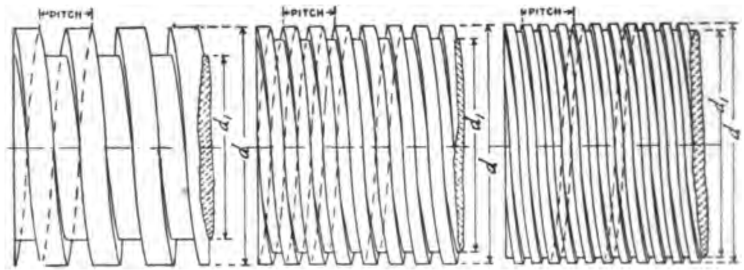


FIG. 297. FIG. 298. FIG. 299.

smaller will the diameter be at the bottom of the threads, therefore the weaker the screw becomes; so, to obviate this, when large pitches are required, screws with two or more threads of the same pitch running parallel to each other are used. Fig. 298 shows a screw with a **double thread,** and Fig. 299 one with a **treble thread,** both of these having the same *diameter* and *pitch* as the *single threaded* one. The pitch of two adjacent threads (or the pitch of the screw divided by the number of threads) is called the **divided pitch,** and it should be evident that *the smaller this pitch the greater will be the total shear resistance and working surface of the screw.*

132. Square Threads versus Vee Threads.—If the tensile load on a vee threaded screw in the direction of its length be represented by T in Fig. 300, then N and B may represent the normal pressure on the threads (also a measure of the friction), and the bursting force on the nut, respectively. From this triangle of forces it is evident that the larger the angle between the threads the greater will the bursting action be and the larger the amount of friction; and, of course, the converse is true. Indeed, when the angle decreases to the vanishing point and the sides of the threads become parallel, we have the case of the square threaded screw, where the axial tension in the screw is nearly equal to the normal pressure on the threads. Hence with this screw there is no bursting action on the nut, and for these reasons *the square thread is preferred for driving purposes or transmitting motion.* But for the same depth of thread the *vee thread* has about twice the amount of material resisting the shearing action at the root of the thread *ab* than a screw with square threads, so, even if square threads could be produced as cheaply as vee threads, the latter would be preferred in ordinary cases where strength is the main consideration. The buttress thread, Fig. 233, as has been explained, to a large extent combines the advantages of the vee and square threads for certain purposes.

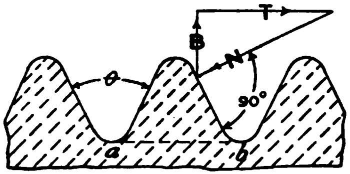


FIG. 300.

The following table should be helpful; it shows what working stresses experience has proved to be satisfactory for the cases referred to, but in special cases, where some of the conditions are not quite the same, the young engineer should be guided by what an intelligent grasp of the points brought out by theory and experiment should teach, if accepted working stresses for the particular case are not available; and in this connection, the tables in the following article should be instructive. Probably, in actual practice bolts are, more often than is known, strained much beyond what strict theory would sanction; indeed, Kirkaldy was of opinion that "screwed bolts are not necessarily injured although strained nearly to their breaking point." But we have seen that bolts in joints of cylinders and pipes, or of other vessels under internal pressure, are usually subjected to a stress from screwing up considerably in excess of that due to the pressure, and, as this excess is an uncertain and unknown quantity, the calculated stress on the bolts due to the pressure should be kept low. Of course in joints that are often broken, such as for manhole covers and like fittings, the stress at the root section should very little exceed 2000 lbs. per sq. inch.

133. Working Stress of Bolts and Studs at Root Section [1] Face Joints.—

		Steel	Iron
Largest diameters of Bolts and Studs	$f =$	6000	4800
Under $\frac{7}{8}''$ diameter	$f =$	(4500 to 3000)	(3600 to 2400)
Ordinary Marine Practice	$f =$	5000	4000
Cylinder under 10″ diameters	$f =$	2500	2000

For Rougher Joints, with packing which must be compressed to make the joints tight, to be on the safe side, *the above values of f should be halved.*

EXERCISES.

Designing, etc.

1. Explain how the strength of a screw is influenced by the pitch of its threads.
2. In a certain hydraulic press the whole load of 100 tons is taken on two steel bolts, and the working stress at the root section of the threads has been fixed at 6000 lbs. per sq. inch. What size should the bolts be? and what pitch of the threads would you recommend? Bearing in mind that the material is steel, would you elect to use plus threads? If so, why?

Drawing Exercises.

3. Make drawings of the following: Whitworth bolts, $\frac{1}{2}''$, $\frac{3}{4}''$, and 1″, showing three views of each. Full size.
4. Make working drawings of a capstan nut (Fig. 287A) for a 3″ bolt. Show it fitted with suitable pin. Scale full size.
5. Set out a loose collar for a 3″ shaft with a *cupped* set-screw.

Sketching Exercises.

6. Make a sketch showing the true form and proportions of the standard Whitworth thread.
7. Make sketches of the following:—a tee-head bolt, hook bolt, eye bolt, stud, forcing screw, adjusting screw, set-screw with saddle piece, flanged nut, cap nut, thumb nut.
8. Sketch a Lewis bolt, also a Rag bolt for use with keys.
9. Make a sketch of a *loose collar*, and show an application of it.

[1] Refer to Art. 148 in "Hydraulic Pipe Joints."

CHAPTER XIII

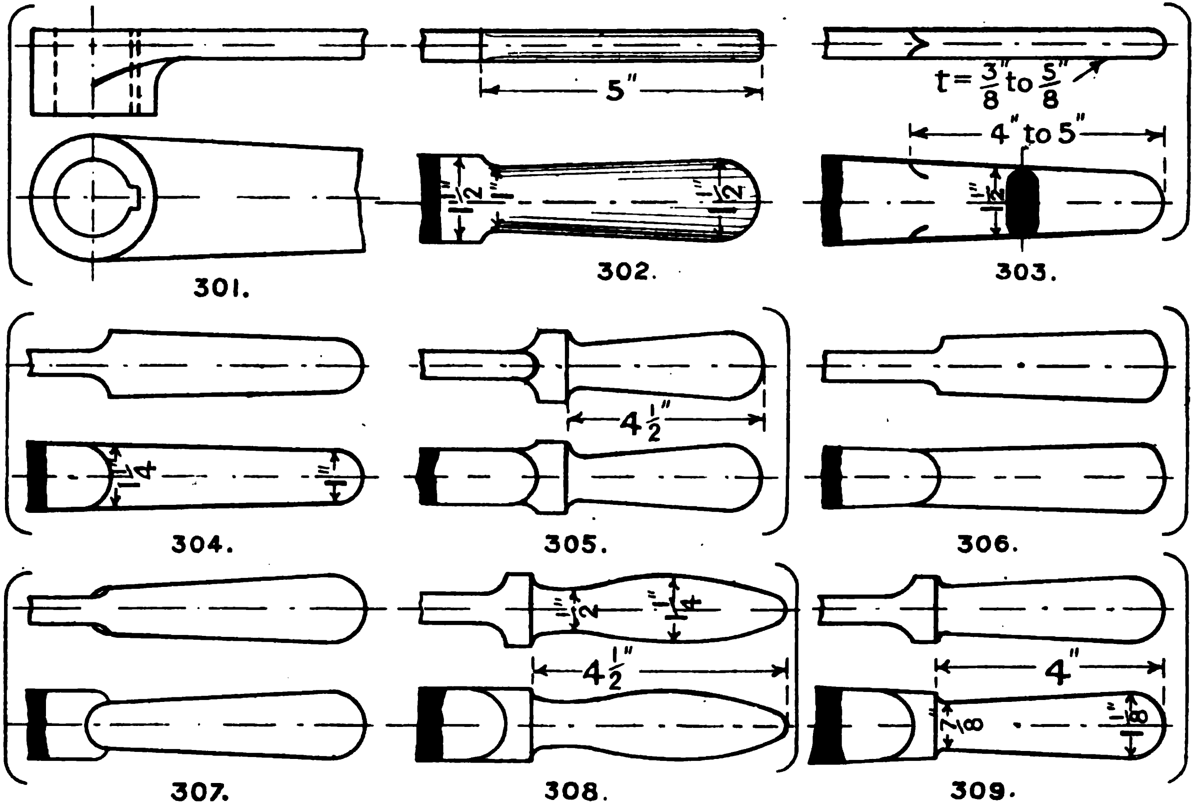
MACHINE HANDLES, ETC.

134. Among the minor details of machines that deserve careful attention in the drawing office are the handles; for few things add more to the convenience in working, and to the general attractiveness of a machine, than suitable and neatly designed handles. They are fittings which vary very much in form, even when used for the same type of machine, so therefore we shall be justified in examining a number of those most frequently met with, and remarking on their good or faulty features. Now, although we have no standard of comparison, we may admit that an *ideal handle* should be **strong** and **durable, simple** and **cheap,** be **pleasant to manipulate** and **elegant in appearance.** As these requirements are somewhat conflicting, usually the best compromise is aimed at. Commencing with **hand levers,** or lever handles, the least expensive form is the flat type, one of which is shown in Fig. 303; it is strong and durable, looks very well, is comfortable to handle, and needs no machining. For finished work it is filed up and polished, or ground, but for rough work it is used direct from the forge or trimmed with the file. Another example of this type is shown in Fig. 302; it is perhaps somewhat more pleasing to look upon, and a better hold can be got when the pull is in the direction of its length, but it is not quite so cheap to forge. A form of handle largely used is shown in Fig. 304; it is easily made, but is clumsy in appearance, although comfortable to handle. Figs. 305 and 309 show a type of lever handle which is more commonly used than any other, being suitable for any size work. It is easily made and cheaply turned and milled, if for finished work; and it has a neat appearance and is strong. The end may be hemispherical, as in Figs. 307 and 309, or somewhat flatter, as in Fig. 306, or flat with rounded corners; but when made flat in this way it is not quite so pleasant to handle. The bead on the latter is finished with a spring tool; it gives to the handle a distinctive appearance but somewhat increases its cost. Fig. 301 is an elegant form, designed to fit the palm of the hand, but it is not often met with now, except in small work, as it is more expensive to turn, requiring, if not finished by hand tools, a special former-plate attachment to the slide-rest of the lathe. In Figs. 306 and 307, the shoulder is dispensed with, but the latter requires some handwork to finish it after turning. Fig. 301 shows a typical boss-end suitable for all the handles from 302 to 309. A very useful form of balanced handle suitable for horizontal shafts is shown in Figs. 310 and 311 (they are largely used on **machine tools**); the boss is shown round, but sometimes it is made square. Figs. 312 and 313 show a convenient form of **Tightening Handle,** which is largely used for **gripping** purposes, such as for holding the back-centre bolt of lathes. The angle θ depends upon the amount of clearance required for the hand.

Small handles that take the form of a **star-ball wheel,** Figs. 314 and 315, are occasionally met with; usually they are either cast, or are stampings. The **Capstan Handle,** Figs. 316 and 317, is another example of this kind, suitable for larger work, where both hands are used to manipulate it. This type of handle is also used to give motion to the tables of hand-planing machines.

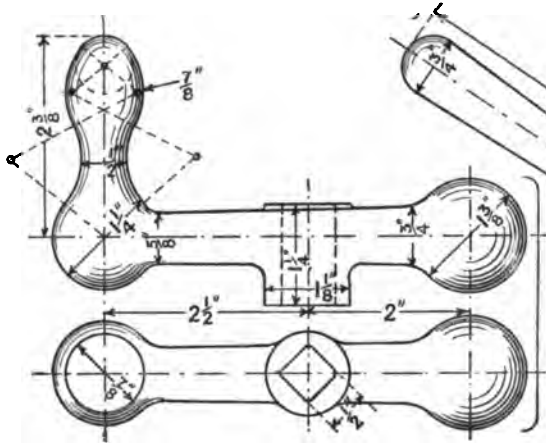
An important type of handle is shown in Figs. 318 and 319; it is largely used for **hand-feed screws**, such as are used for slide-rests. The boss part may be made to the proportions shown in terms of *d*, the diameter of the square end of the screw; the

LEVER HANDLES.

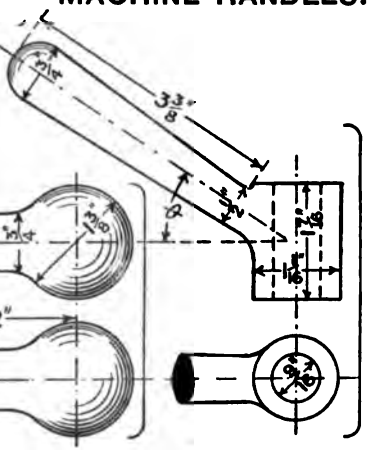


dimensions shown on the handle itself are suitable for a large range of sizes. A handle of another shape is shown in Fig. 320; but the remarks made in connection with Fig. 308 equally apply to this. These handles are shown riveted, but they are sometimes stamped from the solid.

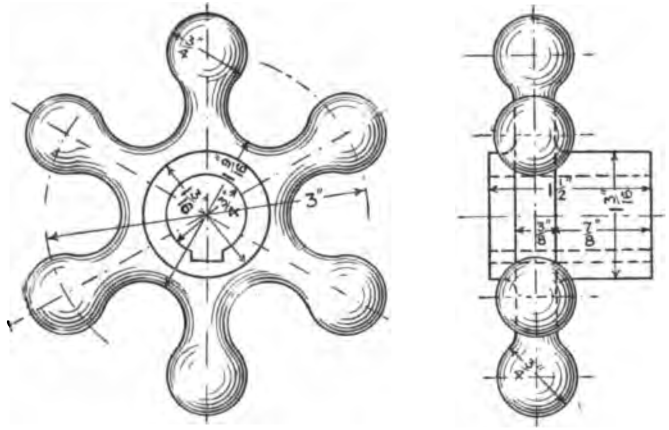
MACHINE HANDLES. DIFFERENT TYPES.



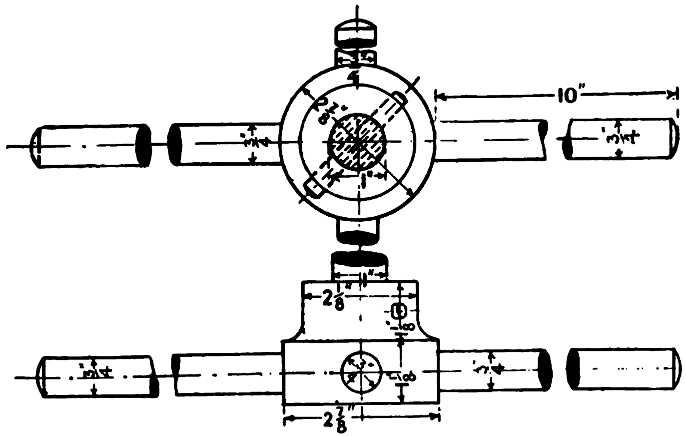
FIGS. 310, 311.—Balanced type.



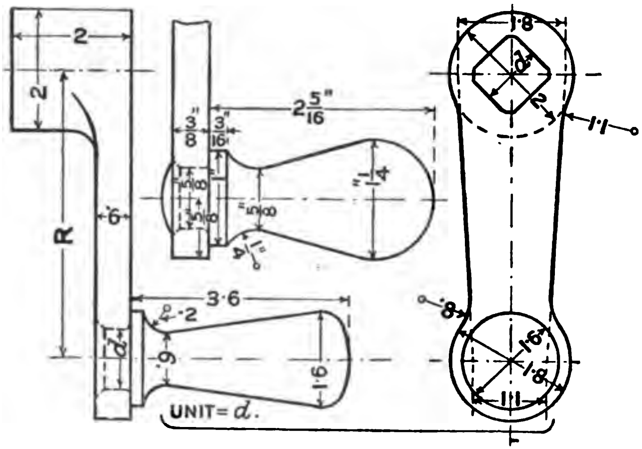
FIGS. 312, 313.—Tightening handle.



FIGS. 314, 315.—Star-ball type.



FIGS. 316, 317.—Capstan type.



FIGS. 318, 320, 319.—Unbalanced types.

Ordinarily, the square, hexagonal, or D-shaped holes in the bosses are slotted and sometimes finished with a drift; but there are now *special drilling machines* used by some makers, by means of which these holes can be accurately drilled out.

EXERCISES.

Drawing Exercises.

1. Show two views of the handle in Figs. 318 and 319, making d of the square hole $\frac{3}{4}''$. Scale full size.
2. Set out the handle in Fig. 320 to fit a square end whose d is $\frac{5}{8}''$.
3. Make drawings of the tightening handle, Figs. 312 and 313, making θ 40°.
4. Draw the two views of the star-ball handle, Figs. 314 and 315, full size.
5. Draw plan and elevation of the capstan handle, Figs. 316 and 317.

Sketching Exercises.

6. Sketch (*a*) a handle suitable for a brake lever, the neatest and cheapest form you are acquainted with; (*b*) a handle suitable for a stop valve.
7. Sketch a machine handle of the form shown in Figs. 310 and 311. What is the use of the ball end?
8. Make a sketch of a handle suitable for the feed screws of a slide-rest.

CHAPTER XIV

PIPES AND PIPE CONNECTIONS

PIPES used by the engineer for the conveyance of steam, water, gas, oil, and other fluids, are made of various metals, including steel, wrought iron, cast iron, and copper. The most suitable material to make pipes of for carrying a particular fluid is decided by taking into consideration such points as cost of production, deterioration, liability to fail, possible result of such failure, nature of the fluid, and its pressure and velocity. But these matters, or at least some of them, can be best touched upon as we explain the kinds of pipes and joints in common use.

STEAM PIPES AND JOINTS.

135. Cast-iron Steam-Pipes.—The accuracy, ease, and certainty with which flanges can be cast with the pipe, and faced and fitted together to form sound joints, and their comparatively low cost, have led engineers in the past to almost universally make use of them for the conveyance of steam; but in recent years pressures have been creeping up, and with them the thickness and weight of these pipes, which have made their disadvantages more apparent, and at each stoppage when in use there is much loss in

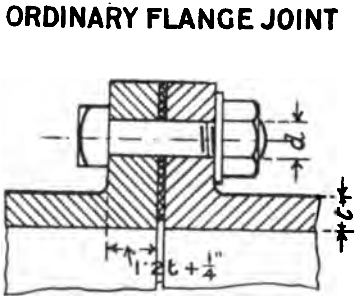


FIG. 321.

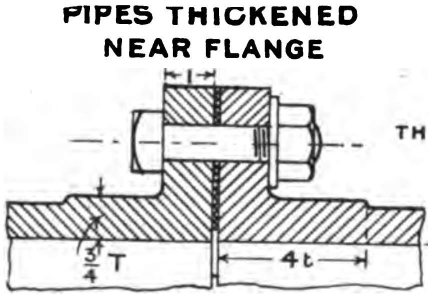


FIG. 322.

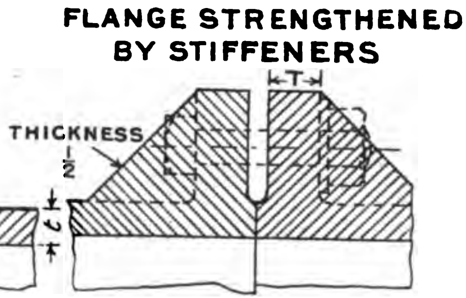


FIG. 323.

re-heating. Moreover, the metal being brittle, it is more liable than other available materials to fail from shock, particularly that due to water-hammer action, and for these reasons cast iron is not used for pressures of over 90, or at the outside 100 lbs., per square inch above atmosphere. Fig. 321 shows the ordinary faced flanged joint.[1] The bolts used have a good deal of strength

[1] For particulars of the standardized proportions of flanges, refer to the author's "Machine Design, etc.," p. 201.

in excess of what is required to resist the internal pressure on a section of the pipe, as they have to maintain a sufficient compression to keep the packing tight and to take the load which may come upon them due to the bending of the pipe between its supports. Their minimum diameter is $\frac{1}{2}''$, used for pipes under 1″ diameter, and the sizes increase with the bores to $1\frac{3}{4}''$ for 24″ bore, and, their maximum distance apart, centre to centre, is about seven or eight diameters. Fig. 322 shows how, for somewhat higher pressures, the metal of the pipe near the flange is thickened to strengthen the connection with it. Fig. 323 shows how the flanges for the highest pressures and largest sizes are still further strengthened by stiffeners between the bolts.

136. Copper Pipes, etc.—The ductility of copper and the ease with which it can be bent or set to any required form, have led to its being largely used for feed and drain pipes for every class of steam engine, and for steam pipes also in marine practice. **Solid drawn copper** pipes are to be had up to about 4″ in diameter, but their uniformity in strength and thickness cannot be absolutely depended upon; larger pipes are made from plates bent to shape and brazed, the flanges being of gun-metal of the shape shown in Fig. 324 for **small sizes and low pressures.** The hole in the flange is slightly countersunk, making a recess for the spelter one on the inner side, and the end of the pipe is swelled to fit the other side before the flange is brazed on. Fig. 325 shows the flange with a strengthening ring B, which gives the flange a better hold, and Fig. 326 shows how copper pipes of 12″ diameter and over are fitted with flanges by **riveting and brazing.** Many disastrous explosions of brazed pipes have happened, and this has led to the practice of fitting important pipes with **bands** brazed round at short intervals, the joints of which are arranged to miss that of the body of the pipe. In Fig. 327 is shown a joint with loose flanges used for rather low pressures; the steel or wrought-iron flanges, if solid, are placed loosely on the solid drawn pipe, which is afterwards flanged over, as shown, or the flanges are made in two or more pieces with overlapping joints. In Pope's flanged joints, angle rings are brazed on the pipes against which the loose flanges are fitted. But iron or steel flanges must not be brazed on to copper pipes, or gun-metal flanges on to iron or steel pipes, on account of their different coefficients of expansion.

ORDINARY COPPER PIPE WITH GUNMETAL FLANGES

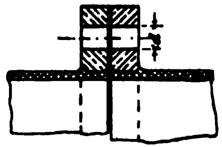


FIG. 324.

COPPER PIPE WITH STRONGER FLANGES

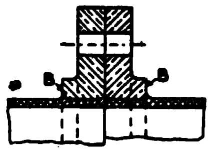


FIG. 325.

COPPER PIPE WITH BRAZED AND RIVETED FLANGES

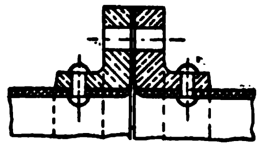


FIG. 326.

COPPER PIPES WITH LOOSE FLANGES OF STEEL OR WROUGHT IRON

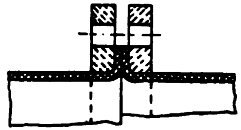


FIG. 327.

137. Wrought-iron and Steel Pipes, etc.—Fig. 328 shows the simplest form of joint for wrought-iron pipes. The ends of the pipes are prepared (or flanged) by working or forming a plain flange, but as this operation slightly reduces the thickness of the flange, and the amount taken off in facing still further reduces it, the joint becomes deficient in rigidity and may prove leaky under any but low pressures. In Fig. 329 is shown a joint formed by solid drawn pipes with **electrically welded flanges,** which make a capital job, with all the good points of cast-iron pipes with the exception of cost, which *at present* is somewhat high, but the ductility of the material makes it much superior in safety to the latter. Fig. 330 shows an excellent joint; the pipe (up to 6″ diameter) is **solid drawn mild steel,** and the **cast-steel flanges** are screwed on and faced. For sizes over

12″ the pipes are often welded and fitted with **riveted flanges,** as in Fig. 331; a cover strip is often fixed over the welded joint of the pipe on the outside as a safeguard, but to be of any real use these should be so proportioned and fitted that

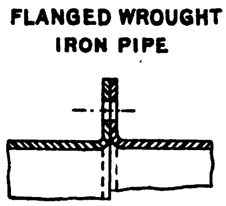


Fig. 328.

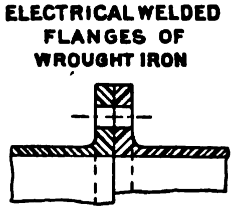


Fig. 329.

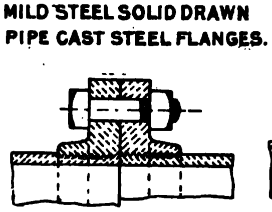


Fig. 330.

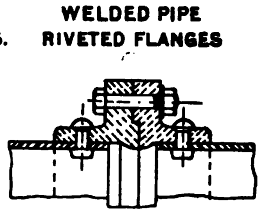


Fig. 331.

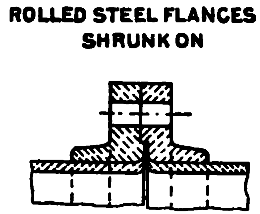


Fig. 332.

they would hold the pipe together independently of the weld. Fig. 332 shows an admirable joint suitable for very high pressures; the **flanges of rolled steel** are shrunk on the pipes of solid drawn steel, a short length of the latter left projecting is spun out or *peened* to expand it over the rounded corner of the flange. The flanges are then faced with a spigot on one and a corresponding recess in the other, so that the packing may not be blown out,[1] and the position of the packing is such that it stops leakage between the flanges, also between pipe and flange.

138. Circumferential[2] (or Longitudinal) Strength of Thin Pipes, Boilers, or other Cylindrical Vessels (subjected to internal pressure), or strength to resist rupture along lines parallel to the axis.

Let D = internal diameter of pipe or cylindrical boiler in inches.
P = working pressure of steam (or other fluid) in lbs. per square inch.
f_t = ultimate strength of material in lbs. per square inch.
F_s = factor of safety.
η = efficiency of joints, if any.
t = thickness of the metal.

Now, let us examine the condition of equilibrium of any section of a cylinder of 1″ breadth made by a diametral plane AB. The total fluid pressure acting vertically upwards will be PD, and this is balanced by the resistance to rupture of the metal at A and B, represented by $2tf_t$. Hence—

$$PD = 2tf_t \quad \text{. . . . . (32)}$$

Or, for working conditions, taking account of the joints—

$$PDF_s = 2tf_t\eta \quad \text{. . . . . (33)}$$

And the following example gives an application of the equation.

139. Example.—The internal diameter of a steel steam pipe, double riveted, is 24″, the efficiency of the joints being

[1] But, on the other hand, the pipes have to be forced apart when being repaired or taken down.
[2] For strength of ring joints, or transverse strength, and strength of spherical vessels, refer to author's "Machine Design, Construction and Drawing," p. 200.

70 per cent., the factor of safety 6, steam pressure 140, and the ultimate strength of the plates 28 tons per square inch. What should the thickness of the plates be?

By Eq. 33, $$\mathrm{PDF}_s = 2tf_t\eta$$

Therefore, by transposition

$$t = \frac{\mathrm{PDF}_s}{2f_t\eta} = \frac{140 \times 24 \times 6}{2 \times 28 \times 2240 \times \frac{70}{100}} = 0{\cdot}229''$$

or, say, $$t = \tfrac{9}{32}''$$

140. Steam-Tubing and Fittings.—A great deal of the small pipework (3″ and less) about a steam plant, both for water and steam, usually consists of wrought-iron steam tubing, the pipe connections in common use being shown in Figs. 333 to 338, which speak for themselves. The joints are made with a thick paint of red lead and oil, the pipe being screwed well home into each fitting. Although the pipes are easily bent and set when hot to any desired form, satisfactory jobs free from serious flattening of sections at bends, leaky joints, and other defects can only be turned out by men who have had a good deal of experience in such work. The **Nipple Connection**, Fig. 336, can only be used when the reduction in the size of the hole due to the nipple is of no consequence. In the **Perkin's Joint**, Fig. 337, the socket is screwed right- and left-handed, and one pipe-end is *conical inside and out*, forming

PIPE CONNECTIONS (WROUGHT IRON.)

BULGED SOCKET

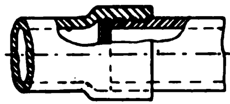


FIG. 333.

ORDINARY SOCKET

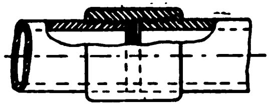


FIG. 334.

HEXAGONAL SOCKET

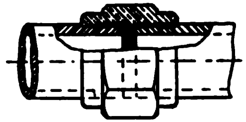


FIG. 335.

NIPPLE CONNECTION

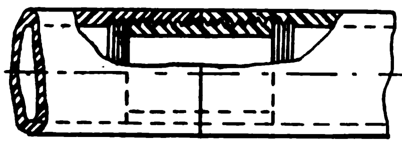


FIG. 336.

PERKIN'S JOINT

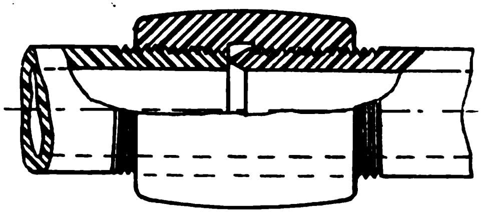


FIG. 337.

PERKIN'S JOINT WITH COPPER WASHER

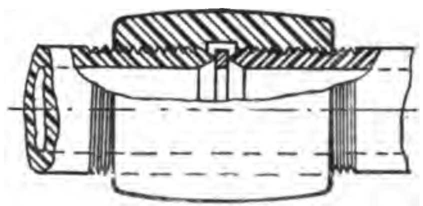


FIG. 338.

a sharp edge, which, when forced on to the flat end of the other pipe by screwing up the socket or coupler, forms a perfect

metal-to-metal joint, through which nothing can escape.[1] Fig. 338 shows a modification of the joint, used when the joint is so awkwardly placed that a large pair of tongs cannot be used to pull it up. A flat soft copper washer is placed between the two pipe-ends (which are in this case both coned), and very little turning effort on the socket makes the joint.

141. Jointing.—Steam flanges are jointed with either asbestos board, rubber asbestos, wire gauze and red lead, corrugated copper rings, very thin (not over $\frac{1}{16}''$) washers of indiarubber, lenticular packing,[2] rings of small lead wire, or string smeared with red lead. But steam pipes, subjected to great variations of temperatures as they are when superheated steam is used, are most difficult to keep tight, and the joints for these are often made without packing, the faces being scraped true and smeared with oil.[3]

SPECIAL JOINTS.

LENS JOINT.

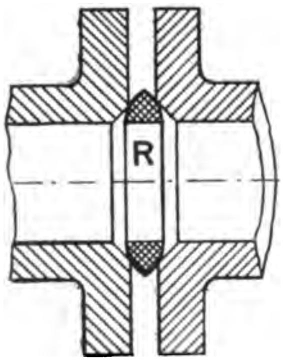


Fig. 339.

UNION JOINT.

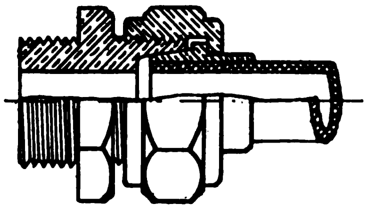


Fig. 340.

High-pressure water pipes are usually jointed with rubber insertion, containing fine wire gauze, but unfaced flanges for water pipes are jointed by bolting between them a wrought-iron ring, wound with rope covered with red lead or tar, the space between flanges being tightly caulked with iron borings to form a rust joint.

142. Special Joints.—A convenient form of **flexible joint,** capable of adjusting itself to a small change in the relative positions of the two pipes, is shown in Fig. 339; it is called the **Lens Joint.** The ring has spherical surfaces, and is made of gun-metal.

Fig. 340 shows one form of a **union joint** made of brass and used for small (usually $\frac{3}{4}''$ and under) brass and copper pipes, where it may be necessary to frequently make and break the joint; the figure should speak for itself.

143. Use of Expansion Joints.—In arranging steam or hot-water pipes the greatest care must be taken, particularly with the former, to provide for the alterations of length and form due to varying temperatures, without allowing the pipes and fittings to be subjected to any but the smallest straining actions.[4] The amount of expansion per foot run can be readily calculated from the following table, due to Kempe:—

TABLE 4.—Expansion Coefficients for Pipe Metals (Kempe).

Metal.	Coefficient.	Tested between	Metal.	Coefficient.	Tested between
Cast iron	0·00000618	32°–212° F.	Wrought iron	0·00000895	32°–572° F.
Steel	0·00000600	32°–212° F.	Copper	0·00000955	32°–212° F.
Wrought iron	0·00000656	32°–212° F.	Copper	0·00001092	32°–572° F.

[1] The author, in testing pipes fitted with such joints, has often, at pressures of some 4000 or 5000 lbs. per sq. inch, squeezed water through the pores of the metal, whilst the joints have remained tight.

[2] Taking the form of an annulus of soft copper, usually made by cutting rings from thick solid drawn copper pipes, the sections of the rings being grooved in various ways, so that the sharp edges when in contact with the flange bases easily make a metal-to-metal joint. Some firms, such as the Combination Metallic Packing Co., make a speciality of these, and stock a variety of sections and sizes; but this type of packing should only be used when the flanges are very strong and rigid.

[3] This is the best practice for very high-pressure steam pipes.

[4] The copper exhaust pipes for petrol engines are frequently made with sharp elbows, where they should be arranged with easy bends.

Thus, with steam at 240 lbs. per sq. inch, with a range of temperature from 32° to nearly 380° F., the expansion per 100′ would be, for cast iron or steel, $0·000006 \times 380 \times 12 \times 100 = 2·756''$, say 3″. In some cases the range may be, for pipes beyond the superheater, some 600° F. Then, with wrought-iron pipes (taking the coefficient at 0·000009) we get $0·000009 \times 600 \times 12 \times 100 = 6·48''$, say $6\frac{1}{2}''$ expansion (above the length when cold) per 100′.

It is usual in good boiler practice to make the branch pipes (which connect the boiler stop valves to the main steam-pipe), at least 12′ long, to give the necessary relief.

144. Pipe Hangers and Bearers.—Fig. 341 shows a simple way that heavy pipes are sometimes supported in, which is obviously defective, as any movement of the pipe in the direction of its length would probably cause the clip to heel on one of its supports, and slightly lift the pipe out of position. The hanger should be fitted with a roller, as shown in Figs. 341A to

PIPE HANGERS AND BEARERS.

DEFECTIVE SUPPORT. EFFICIENT SUPPORT. GROUND ROLLER SUPPORTS OR BEARERS.

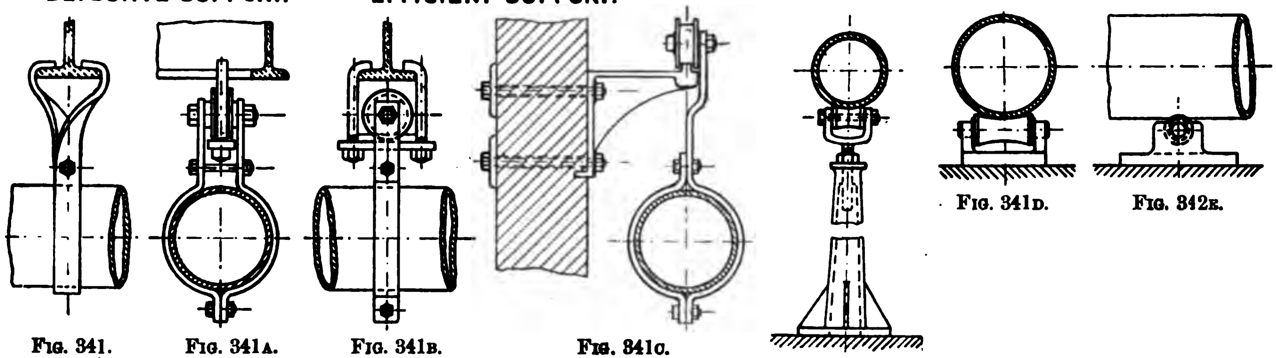


FIG. 341. FIG. 341A. FIG. 341B. FIG. 341C. FIG. 341F. FIG. 341D. FIG. 342E.

341C. Steam pipes should not rest on the ground, but be supported at suitable intervals by cast-iron roller-bearing blocks resting on a bed of concrete or stone, as shown in Figs. 341D and 341E, or if some distance from the ground, by a roller standard, Fig. 341F.

145. Expansion Joints, etc.—We have explained under what conditions expansion joints become necessary, and we have in Fig. 342 a length of copper pipe (or of lap-welded or weldless steel pipe, with riveted flanges) bent into the form of a horseshoe, which may take the place of a short length in a pipe; it offers little resistance to the ends A and B being moved closer together or further apart, by expansion and contraction. Fig. 342A shows a *loop* arranged for the same purpose; but these expansion joints are only reliable when they are very little stressed by such straining actions, particularly when made of copper, as that metal has a **low elastic limit,** and **less spring** than wrought iron or mild steel. The same remarks apply to the *cushion expansion joint* shown in Fig. 342B, and also to the *corrugated expansion joint*, Fig. 342C (which is a development of the latter); these can only be safely used to serve ***very short*** lengths of pipe that for some reason or other have to be straight. The difficulty with these joints is that there is

sure to be some one part or corrugation that is a little weaker to resist compression or tension than the others, with the result that it takes up all the work, and the joint ultimately fails[1] at that part, and when two or more such joints are used in one length of pipe,

EXPANSION JOINTS.

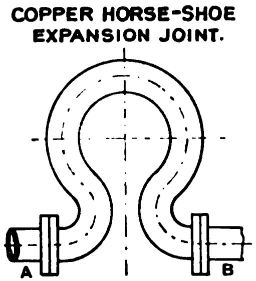
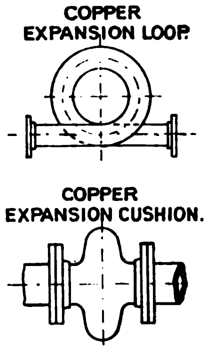


FIG. 342.



FIGS. 342A, 342B.

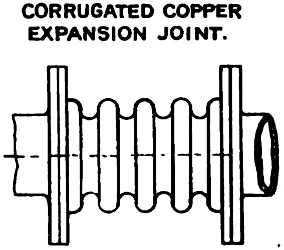


FIG. 342C.


FIG. 342D.—Messrs. Crane & Co.'s malleable cast-iron expansion bends.

one joint alone may take up nearly the whole expansion; for this reason more than one such joint per length of pipe should never be used. The battery of expansion bends shown in Fig. 342D, is manufactured by Messrs. Crane & Co. The three bends have a total sectional area about equal to that of the main pipe,[2] their smaller diameter giving greater flexibility. On the whole, perhaps by far the most satisfactory expansion joint is the well-known **gland and stuffing-box arrangement** shown in Fig. 343, which should always be fitted with guard-bolts A, B, to prevent the two parts being blown apart, should any movement of either end of the pipe take place. These bolts sometimes are also used as studs for the stuffing box. When the skin of cast iron is removed by machining, the metal quickly oxidizes, so, to prevent rust joints being formed, the working surfaces in the best work are gun-metal, the parts being bushed and sheathed, as shown in Fig. 343.

145a. Drawing Exercise.—Fig. 343. Draw the sectional elevation. A plan and two end elevations. Scale, half full-size.

146. British Standard Pipe Threads.—The Engineering Standards Committee have recommended that the Whitworth thread (Art. 117) should be employed for all iron or steel tubes and *couplers* or sockets; also for tubes made from copper, brass, or similar metal, and for these latter materials where the outside diameters agree, and the thickness of the metal permit, *the same pitches be adopted.* The committee has formulated full particulars of pipe threads for *nominal bores* of $\frac{1}{8}''$ to 18″, which are now known as the

[1] The author gave these joints (the corrugated ones) a trial some years ago, but they were so unsatisfactory that he had to replace them by joints of the gland and stuffing-box type.
[2] Refer to author's "Machine Design, Construction and Drawing," footnote, p. 196.

British Standard pipe threads,[1] and this standardization will doubtless be greatly appreciated by all who have to do with pipe threads from various manufacturers. Table 5 gives a few ordinary particulars relating to pipe threads.

GLAND & STUFFING-BOX EXPANSION JOINT.

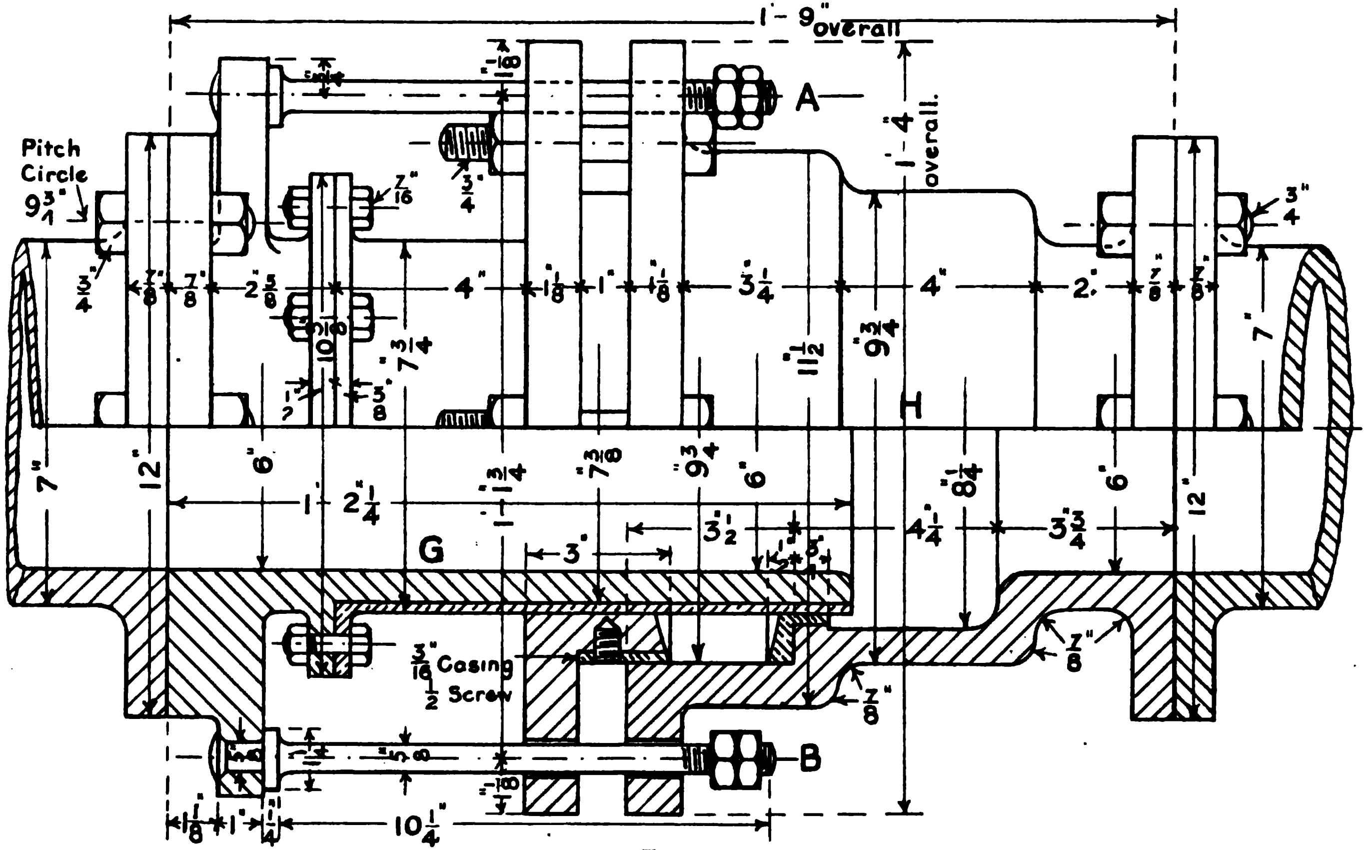


FIG. 343.

[1] The Report issued by the Committee contains their recommendations, and a large amount of valuable information relating to these screw threads, in addition to the Tables. It is published by Messrs. Crosby Lockwood & Son, at 2*s*. 6*d*. net.

TABLE 5.—Pipe Threads.

The Number of Threads per Inch have not been altered by the Standard Committee. They are as follows:—

Nominal bore of tube.	Number of threads per inch.	Nominal bore of tube.	Number of threads per inch.
$\frac{1}{8}''$	28	$1''$ to $6''$	11
$\frac{1}{4}''$ and $\frac{3}{8}''$	19	$7''$ to $10''$	10
$\frac{1}{2}''$ to $\frac{7}{8}''$	14	$11''$ to $18''$	8

The bores advance by $\frac{1}{8}''$ from $\frac{1}{8}''$ to $\frac{7}{8}''$, by $\frac{1}{4}''$ from $1''$ to $4''$, by $\frac{1}{2}''$ from $4''$ to $6''$, and by inches from $6''$ to $18''$.

Diameters of Screwed Part, Core, and of Black Tube.

Nominal bore of tube.	Approximate outside diameter. Black tube.	Diameter top of thread.	Diameter of core.	Nominal bore of tube.	Approximate outside diameter. Black tube.	Diameter top of thread.	Diameter of core.
Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.
$\frac{1}{8}$	$\frac{13}{32}$	0·383	0·337	$1\frac{1}{4}$	$1\frac{11}{16}$	1·650	1·534
$\frac{1}{4}$	$\frac{17}{32}$	0·518	0·451	$1\frac{1}{2}$	$1\frac{29}{32}$	1·882	1·766
$\frac{3}{8}$	$\frac{11}{16}$	0·656	0·589	$1\frac{3}{4}$	$2\frac{5}{32}$	2·116	2·000
$\frac{1}{2}$	$\frac{27}{32}$	0·825	0·734	2	$2\frac{3}{8}$	2·347	2·231
$\frac{5}{8}$	$\frac{15}{16}$	0·902	0·811	$2\frac{1}{4}$	$2\frac{5}{8}$	2·587	2·471
$\frac{3}{4}$	$1\frac{1}{16}$	1·041	0·950	$2\frac{1}{2}$	3	2·960	2·844
$\frac{7}{8}$	$1\frac{7}{32}$	1·189	1·098	$2\frac{3}{4}$	$3\frac{1}{4}$	3·210	3·094
1	$1\frac{11}{32}$	1·309	1·193	3	$3\frac{1}{2}$	3·460	3·344

147. Spigot and Socket Joints.—Cast-iron pipes, used for the conveyance of low-pressure water or gas, which have to be embedded in the ground, are connected by spigot and socket joints, which have a certain amount of **flexibility** at the joints, allowing the pipe to accommodate itself to slight settlements of the earth. The proportions and details somewhat differ, but Fig. 344 shows a typical example of the joint for cast-iron pipes as used by Mr. Bateman at the Glasgow Waterworks, and his proportions for various sizes are shown in Table 6. The joint is made by first driving a few coils of **gasket** or yarn into the socket and then filling the remaining space with lead, which is done by putting a clay band round the outside of the socket and running in the **molten lead**, which, when cold enough, is **caulked** or **stemmed** tightly into the socket, and the clay is removed. The socket is sometimes grooved, as at G, Fig. 345, to better prevent the lead being blown out. Fig. 344A shows a form of **turned and bored** spigot and socket joint; the taper of the bored part is $\frac{1}{32}''$ per inch of length; the joint is made fluid tight by painting the turned parts with red lead or

liquid Portland cement before putting them together with a blow or two to drive the spigot home. The socket is then filled up with cement, as shown.

TABLE 6.—PROPORTIONS OF (LEAD) SPIGOT AND SOCKET JOINT. BATEMAN. (Fig. 344.)

Bore in inches.	Length of each pipe-ft	A	B	C	D	E	F	G	H	T	t
2	6	3	3	$\frac{1}{4}$	$\frac{7}{8}$	$\frac{3}{4}$	$\frac{11}{16}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{1}{2}$	$\frac{3}{8}$
3	9	3	3	$\frac{1}{4}$	$\frac{7}{8}$	$\frac{3}{4}$	$\frac{11}{16}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{1}{2}$	$\frac{3}{8}$
4	9	3	3	$\frac{1}{4}$	$\frac{7}{8}$	$\frac{3}{4}$	$\frac{11}{16}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{1}{2}$	$\frac{3}{8}$
5	9	$3\frac{1}{2}$	$3\frac{1}{2}$	$\frac{3}{8}$	1	$\frac{7}{8}$	$\frac{13}{16}$	$\frac{11}{16}$	$\frac{7}{8}$	$\frac{5}{8}$	$\frac{7}{16}$
6	9	$3\frac{1}{2}$	$3\frac{1}{2}$	$\frac{3}{8}$	1	$\frac{7}{8}$	$\frac{13}{16}$	$\frac{11}{16}$	$\frac{7}{8}$	$\frac{5}{8}$	$\frac{7}{16}$
7	9	$3\frac{1}{2}$	$3\frac{1}{2}$	$\frac{3}{8}$	1	$\frac{7}{8}$	$\frac{13}{16}$	$\frac{11}{16}$	$\frac{7}{8}$	$\frac{5}{8}$	$\frac{1}{2}$
8	9	4	4	$\frac{3}{8}$	$1\frac{1}{8}$	1	$\frac{13}{16}$	$\frac{11}{16}$	$\frac{7}{8}$	$\frac{5}{8}$	$\frac{1}{2}$
9	9	4	4	$\frac{3}{8}$	$1\frac{1}{8}$	1	$\frac{13}{16}$	$\frac{11}{16}$	$\frac{7}{8}$	$\frac{3}{4}$	$\frac{9}{16}$
12	9	4	4	$\frac{3}{8}$	$1\frac{1}{4}$	1	$\frac{15}{16}$	$\frac{13}{16}$	$1\frac{1}{8}$	$\frac{3}{4}$	$\frac{5}{8}$
15	9	$4\frac{1}{2}$	$4\frac{1}{2}$	$\frac{3}{8}$	$1\frac{3}{8}$	$1\frac{1}{16}$	$\frac{15}{16}$	$\frac{15}{16}$	$1\frac{1}{4}$	$\frac{7}{8}$	$\frac{11}{16}$
18	9	$4\frac{1}{2}$	$4\frac{1}{2}$	$\frac{3}{8}$	$1\frac{1}{2}$	$1\frac{1}{4}$	$1\frac{1}{8}$	1	$1\frac{1}{4}$	1	$\frac{13}{16}$
20	9	$4\frac{1}{2}$	$4\frac{1}{2}$	$\frac{3}{8}$	$1\frac{5}{8}$	$1\frac{3}{8}$	$1\frac{1}{4}$	$1\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{1}{8}$	$\frac{7}{8}$
24	12	5	5	$\frac{3}{8}$	$1\frac{3}{4}$	$1\frac{1}{2}$	$1\frac{3}{8}$	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{5}{16}$	1
33	12	$5\frac{1}{2}$	5	$\frac{3}{8}$	2	$1\frac{3}{4}$	$1\frac{1}{2}$	$1\frac{3}{8}$	$1\frac{5}{8}$	$1\frac{5}{16}$ to $1\frac{9}{16}$	1 to $1\frac{1}{4}$

TABLE 7.—TURNED AND BORED SPIGOT AND SOCKET JOINT. (Fig. 344A).

Bore in inches.	Length of each pipe-ft.	A	B	C	D	E	F	G	H	I	T	t
2	6	3	3	$\frac{1}{4}$	$\frac{7}{8}$	$\frac{3}{4}$	$\frac{11}{16}$	$\frac{5}{8}$	$\frac{3}{4}$	$1\frac{1}{4}$	$\frac{1}{2}$	$\frac{3}{8}$
3	9	3	3	$\frac{1}{4}$	$\frac{7}{8}$	$\frac{3}{4}$	$\frac{11}{16}$	$\frac{5}{8}$	$\frac{3}{4}$	$1\frac{1}{4}$	$\frac{1}{2}$	$\frac{3}{8}$
4	9	3	3	$\frac{1}{4}$	$\frac{7}{8}$	$\frac{3}{4}$	$\frac{11}{16}$	$\frac{5}{8}$	$\frac{3}{4}$	$1\frac{1}{4}$	$\frac{1}{2}$	$\frac{3}{8}$
5	9	$3\frac{1}{2}$	$3\frac{1}{2}$	$\frac{3}{8}$	1	$\frac{7}{8}$	$\frac{13}{16}$	$\frac{11}{16}$	$\frac{7}{8}$	$1\frac{3}{8}$	$\frac{5}{8}$	$\frac{7}{16}$
6	9	$3\frac{1}{2}$	$3\frac{1}{2}$	$\frac{3}{8}$	1	$\frac{7}{8}$	$\frac{13}{16}$	$\frac{11}{16}$	$\frac{7}{8}$	$1\frac{3}{8}$	$\frac{5}{8}$	$\frac{7}{16}$
7	9	$3\frac{1}{2}$	$3\frac{1}{2}$	$\frac{3}{8}$	1	$\frac{7}{8}$	$\frac{13}{16}$	$\frac{11}{16}$	$\frac{7}{8}$	$1\frac{3}{8}$	$\frac{5}{8}$	$\frac{1}{2}$
8	9	4	4	$\frac{3}{8}$	$1\frac{1}{8}$	1	$\frac{13}{16}$	$\frac{11}{16}$	$\frac{7}{8}$	$1\frac{3}{8}$	$\frac{5}{8}$	$\frac{1}{2}$
9	9	4	4	$\frac{3}{8}$	$1\frac{1}{8}$	1	$\frac{13}{16}$	$\frac{11}{16}$	$\frac{7}{8}$	$1\frac{3}{8}$	$\frac{5}{8}$	$\frac{9}{16}$
12	9	$4\frac{1}{4}$	$4\frac{1}{4}$	$\frac{3}{8}$	$1\frac{1}{4}$	1	$\frac{15}{16}$	$\frac{13}{16}$	1	$1\frac{3}{4}$	$\frac{3}{4}$	$\frac{5}{8}$

But recent practice favours a shorter turned part, as shown in Fig. 346, its length being some $\frac{5}{8}'$ to $\frac{7}{8}''$, which much increases the flexibility of the joint.

148. Joints for Hydraulic Pipes.—Many years ago the late Lord Armstrong introduced the simple and efficient joint, Figs. 347 and 348, for high-pressure water pipes, and it is now generally used for hydraulic mains. Fig. 349 shows in detail how the joint is

SPIGOT AND SOCKET JOINTS.

ORDINARY SOCKET & SPIGOT JOINT, MADE WITH LEAD

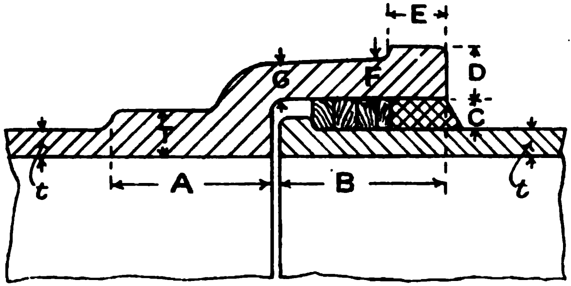


FIG. 344.

TURNED & BORED SPIGOT & SOCKET JOINT, MADE WITH CEMENT

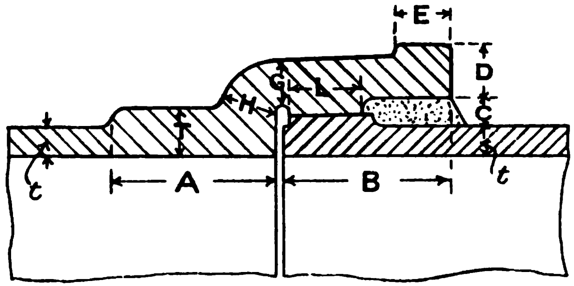


FIG. 344A.

formed. The empirical proportions of the ordinary cast-iron pipe joint in terms of t and d generally used are shown on Figs. 347 and 349. The practice of some engineers is to make the face *ab* of the flange flush, which enables the joint to slightly yield under a lateral load. The usual practice is to subject the metal of the pipe to a **working stress** of 2800 lbs. per sq. inch, and the bolts at the core section to 7700 lbs. per sq. inch, allowing an extra thickness $k = \frac{1}{4}''$ for corrosion, inequalities of the casting, etc., of the former.

But we have seen (Eq. 32, Art. 138) that for a thin[1] cylindrical vessel subjected to internal pressure, we have $PD = 2tf_t$, or $t = \frac{PD}{2f_t}$. Then for, say, a 5″ pipe, and P = 700 lbs. per sq. inch, the thickness—

LEAD JOINT, SOCKET GROOVED

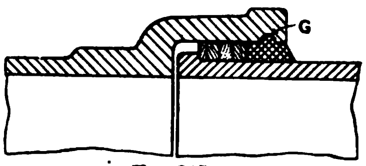


FIG. 345.

TURNED & BORED, SHORT FLEXIBLE JOINT

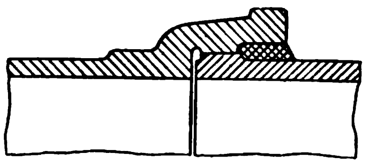


FIG. 346.

$$t = \frac{700 \times 5}{2 \times 2240} + \tfrac{1}{4}'' = 0{\cdot}625 + 0{\cdot}25 = 0{\cdot}875 = \tfrac{7}{8}''.$$

And the above is equivalent to the *simple rule* of $t = \frac{D}{8} + \frac{1}{4}''$ for a pressure of 700 and a stress of 2800 lbs. per sq. inch.

[1] Strictly speaking, these pipes could hardly be called *thin* cylinders, but it is usual to consider them so for this purpose, and any error due to this is well covered by the $\frac{1}{4}''$ allowed.

ARMSTRONG'S HYDRAULIC PIPE JOINT.

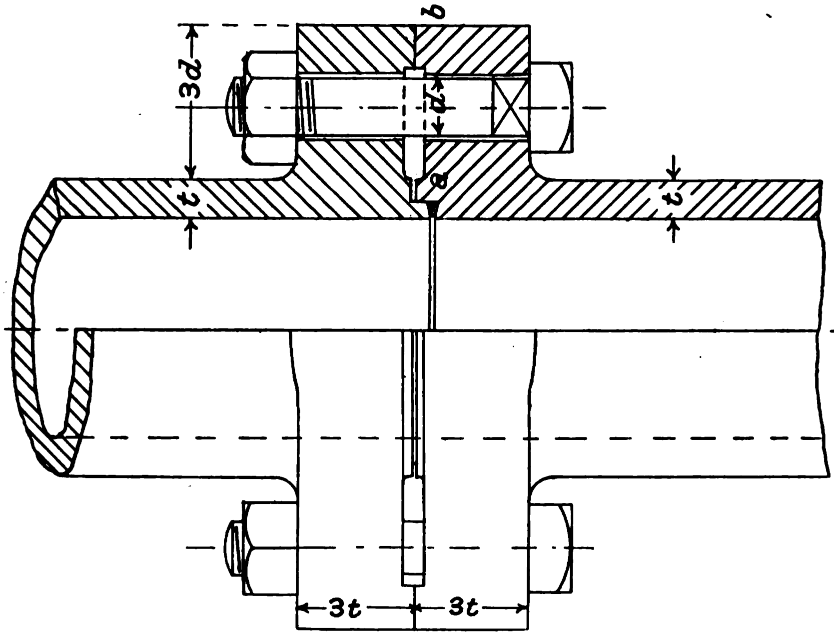


Fig. 347.

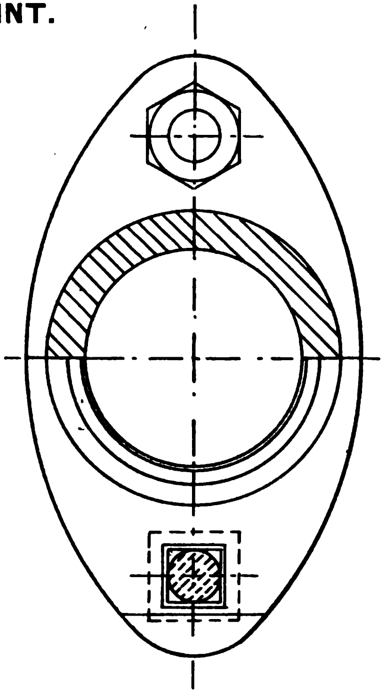


Fig. 348.

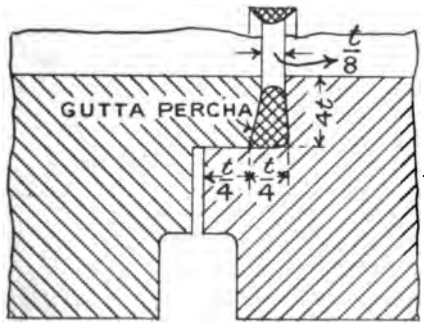


Fig. 349.

HYDRAULIC UNION

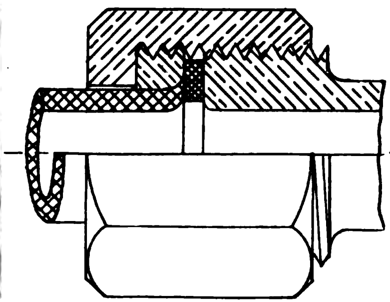


Fig. 350.

HYDRAULIC UNION FOR COPPER PIPES.

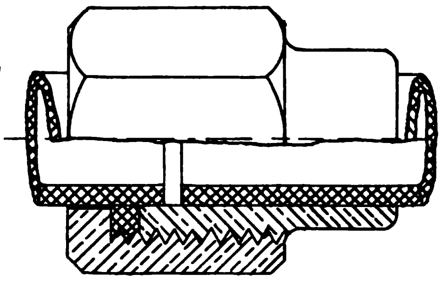


Fig. 351.

Bolts.—Equating the *total* pressure acting on a section of the pipe, to the strength of two bolts, we get $PD^2\frac{\pi}{4} = 2d_1^2\frac{\pi}{4} \times 7700$, where d is the *core* diameter of the bolts.

HYDRAULIC STOP VALVE, HEMP PACKED (Drawing Exercise).

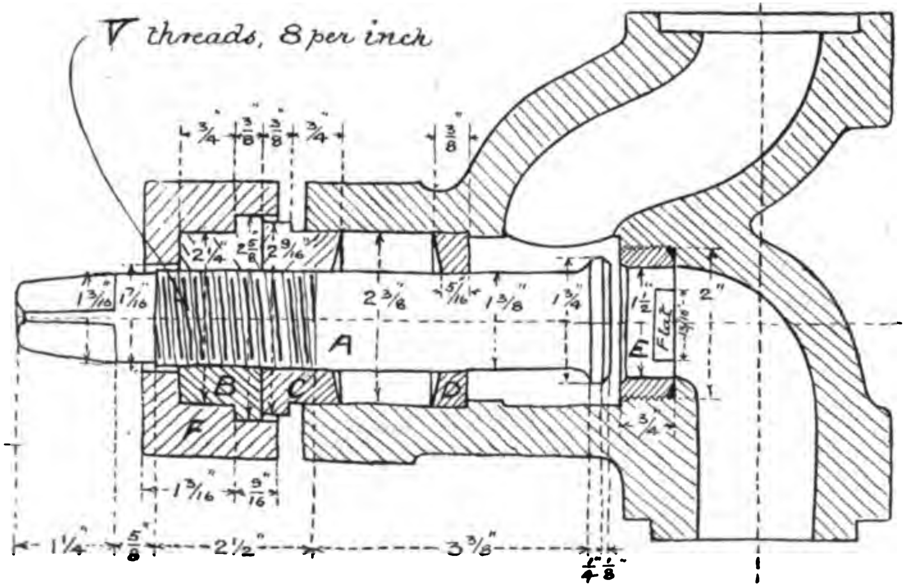


Fig. 352.

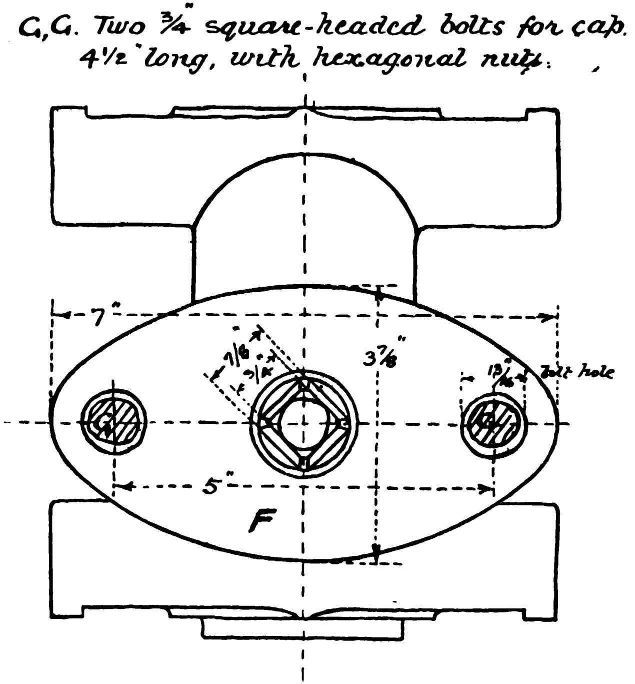
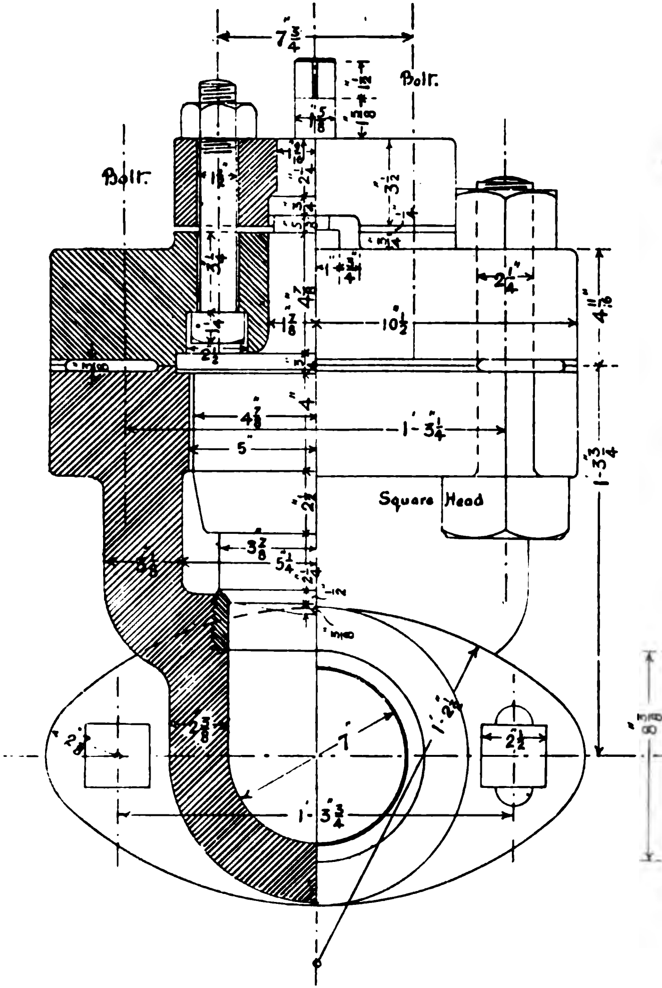


Fig. 353.

Then
$$d_1 = \sqrt{\frac{700 \times 5^2}{2 \times 7700}} = 1{\cdot}066.$$

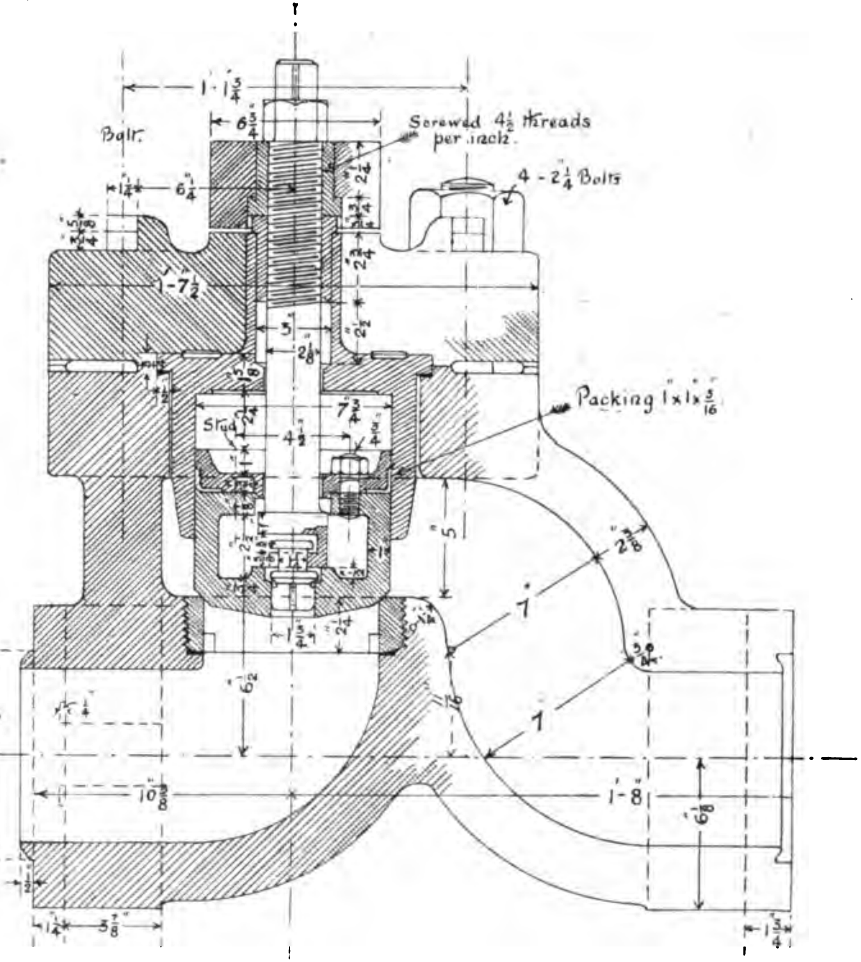
But $d_1 = 0{\cdot}9d - 0{\cdot}05$ (Eq. 25, Art. 117).
$\therefore\ d = \frac{10}{9}(d_1 + 0{\cdot}05) = \frac{10}{9}(1{\cdot}066 + 0{\cdot}05) = 1{\cdot}24$, say $1\frac{1}{4}''$.

HYDRAULIC STOP VALVE, LEATHER PACKED (Drawing Exercise).



Transverse Sectional Elevation.

FIG 354.

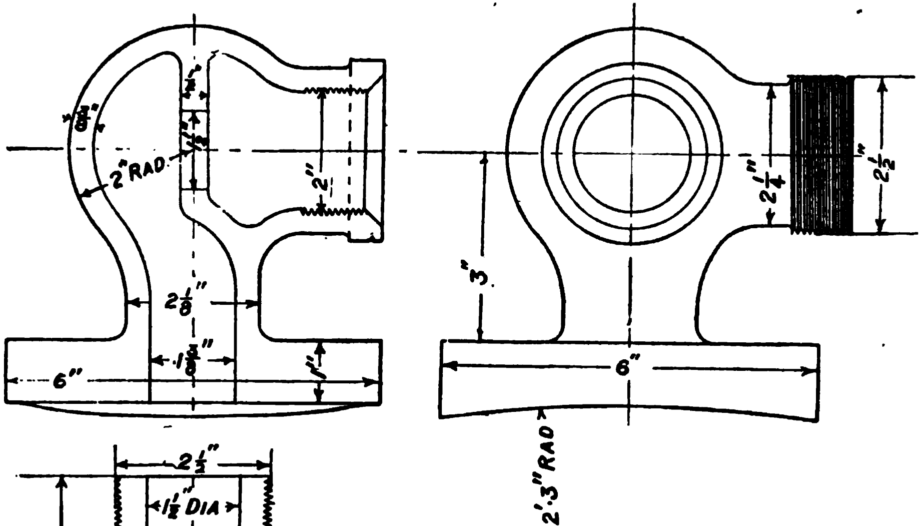


Longitudinal Section.

FIG. 355.

Figs. 350 and 351 show two forms of **Hydraulic Union Joint,** the packing rings being either of leather, gutta-percha, or soft copper.

BODY OF STEAM STOP VALVE (Drawing Exercise).



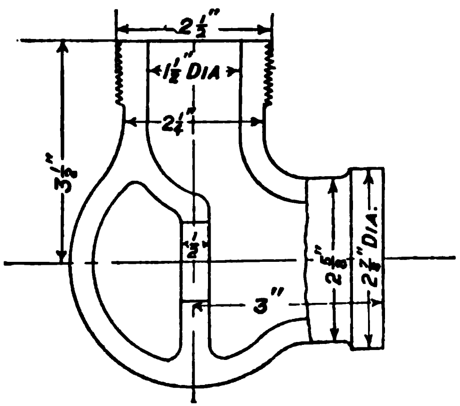


Fig. 355a.

149. Hydraulic Stop Valves (Drawing Exercise).—Two sectional views of an 1½" hydraulic stop valve[1] are shown in Figs. 352 and 353.

Instructions.—Make full-size separate scale drawings of its following details, showing at least two views of each one: Nut B, gland C, bush D, seating E, and cap F.

150. Another Drawing Exercise.—Figs. 354 and 355 are two sectional views of a 7" **hydraulic stop valve,** which, as will be seen, is somewhat more complicated, being arranged for **leather packing,** but the details are so fully dimensioned that an apt student should experience no difficulty in **drawing the two views and adding a plan. Scale, one-quarter full size.**

151. Steam Stop Valve, Body of (Drawing Exercise).—Three dimensioned views of the body of a steam stop valve[2] are shown in Fig. 355a.

Instructions.—Set out the views to a scale of full size. Advanced students should be able to complete the valve by fitting it with a suitable spindle, gland, handwheel, etc.

152. Steam Equilibrium Admission Valve (Drawing Exercise).—In Fig. 355b we have two sectional views of a steam equilibrium admission valve, which were given in the 1907, Stage 2, B. of E. paper. Students were supplied with the following:—

Instructions.—(*a*) Draw a part sectional elevation of the stuffing-box cover and the valve seating, looking in the direction indicated by the arrow K. The part of the elevation to the left of the centre line is to be a section along HG, and the part to the right an external elevation of the valve seating and cover. The valve and the outer casing are to be omitted in this view.

(*b*) Draw a part sectional plan, the part above the horizontal centre line is to be a horizontal section of the valve and seating through EF, and the part below a plan of the cover. The outer casing is again to be omitted.

Scale ½ full size.

Neither dotted lines nor dimensions need be shown.

N.B.—Take the vertical centre line in the direction of the longer dimension of your drawing paper.

[1] This drawing example was given in the B. of E. Exam. Stage 1, 1907. It is arranged for **hemp** or asbestos **packing**.

[2] In the 1907 C. G. Exam. in Mechanical Engineering pattern-makers were asked to make a pattern of this valve.

STEAM EQUILIBRIUM ADMISSION VALVE (Drawing Exercise).

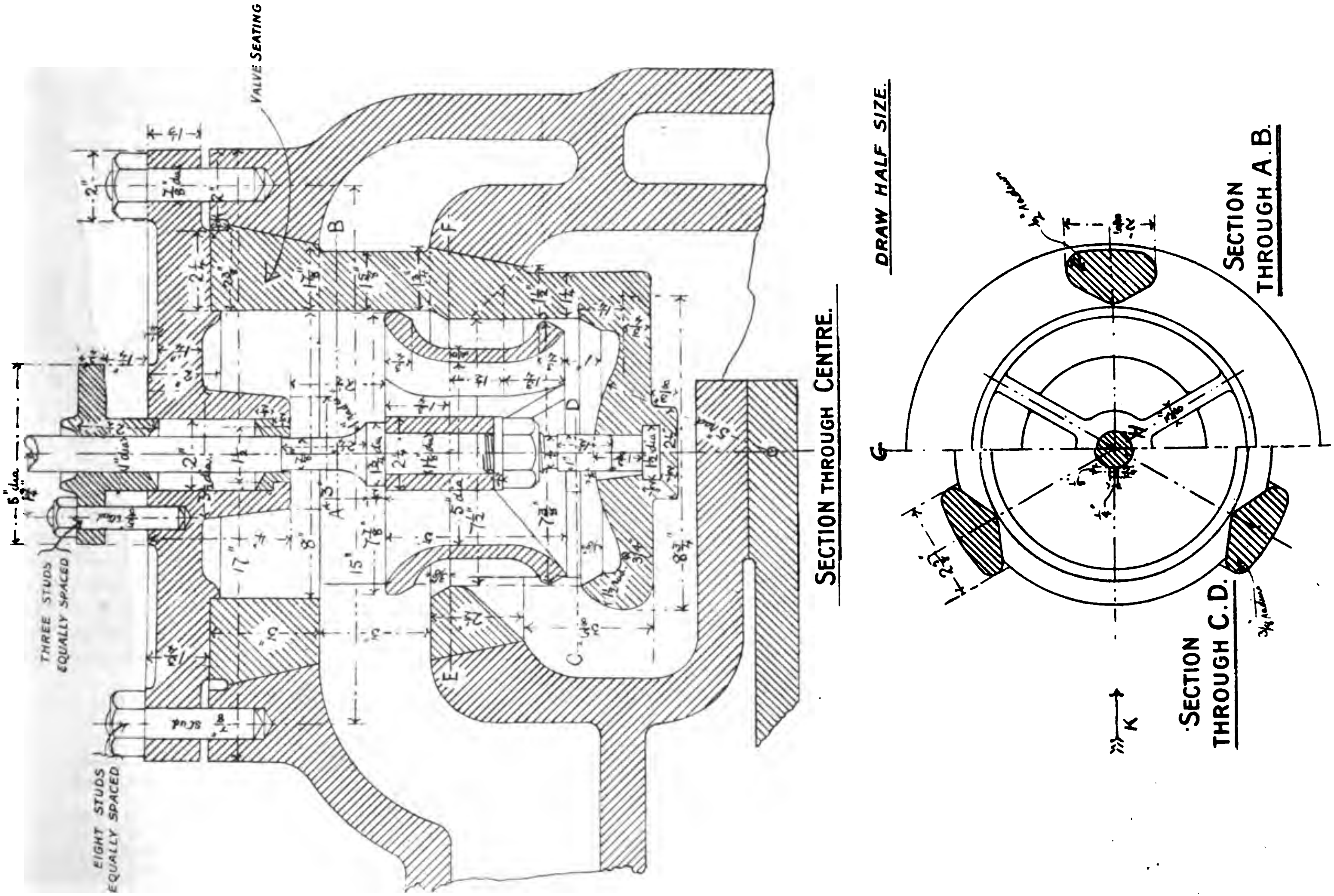


FIG. 355B.

R

153. Thick Cylinder Castings.—When a casting is cooling from its molten condition the heat passes out in the most direct way, that is, in a direction normal to the surface, as shown in the sections of solid and hollow columns, A and B, Fig. 357, and the crystals of the metal (a group of which is shown in Fig. 356) arrange themselves in that direction,[1] so that when two cooling surfaces are at right angles to each other, as at C, Fig. 357, and the passage of heat is equally rapid in both directions, solidification occurs in such a way that confused crystallization results, and a line XY of weakness[2] is produced bisecting the angle, whilst in A and B the lines of crystallization all radiate from the centre, and no interference occurs. The section A (Fig. 358) is a variation of C (Fig. 357), and B is a representative case where sharp angles give lines of weakness, C showing how the use of **fillets** and rounding the corners results in a satisfactory casting. The first very large hydraulic presses were made to raise the gigantic tubes of the

THICK CYLINDER CASTINGS.

Fig. 356.

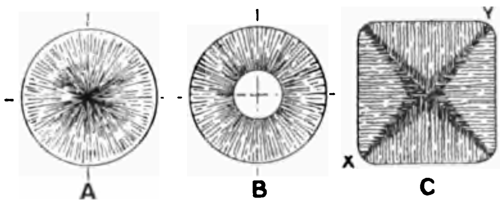


Fig. 357.

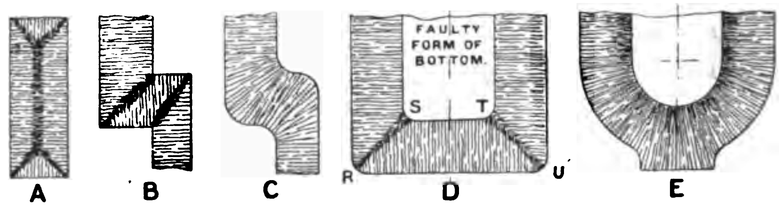


Fig. 358.

Britannia and Conway tubular bridges, the form of the bottom ends of the presses being that shown in the figure D (Fig. 358), but, to the astonishment of the engineers, the bottoms came out, conical in form, the fracture occurring at RS and TU. The same thing occurring when they were made thicker, Mr. Edwin Clerk[3] decided to try the hemispherical form E of uniform thickness, which decision was based on a true knowledge of the cause of failure, and resulted in complete success. But this form is not always convenient in practice, so, as a compromise, the bottom at the inside is usually rounded as in A, Fig. 359, which shows a form that answers well. When the cylinder is so long that a core bar must be passed through its bottom to be supported at that end, the hole is bored and plugged with a slightly tapered plug driven from the inside as in B, Fig. 359. For cylinders over about 12″ diameter a back plate and **U**-leather are sometimes used with the plug, as shown in C, Fig. 359.

The metal used in these castings must not only be strong and tough enough for the purpose, but it must also be of *close texture*,

[1] The molten metal coming in contact with the mould is cooled, and forms a thin lining to the mould, the inner surface of which consists of the tops of crystals (belonging to the *cubic* system) of the metal in groups projecting into the body of the molten metal, and growing in size as cooling and solidification proceed, at the same time arranging themselves so that their longer dimension is at right angles to the cooling surface. For this reason castings (particularly iron ones) always have, or should have, a round or flat *fillet* at the corners or angles.

[2] This weakness is apparently partly due to irregular crystallization, and partly to the separation of the more fusible constituents of the iron and their accumulation in that part.

[3] Mr. Edwin Clerk and his brother Latimer (two of the author's old chiefs) were resident engineers for the bridges under Robert Stephenson, the former of them afterwards becoming so famous in connection with his canal lifts and floating docks.

or the water will ooze through it when under great pressures.[1] And to ensure the castings being sound these cylinders are always cast with a head of substantial volume on the uppermost end[2] in the mould, so as to produce a sufficient fluid pressure on the metal in the mould, and cause the metal to remain fluid long enough to exert this pressure till solidification of the casting occurs.

154. Faults in Designing Cylinders, etc.—When the boiler-maker cuts a manhole in a boiler he is careful to surround it with a ring plate of sufficient section to strengthen the ring of the boiler whose continuity has been destroyed by the cutting of the hole; so

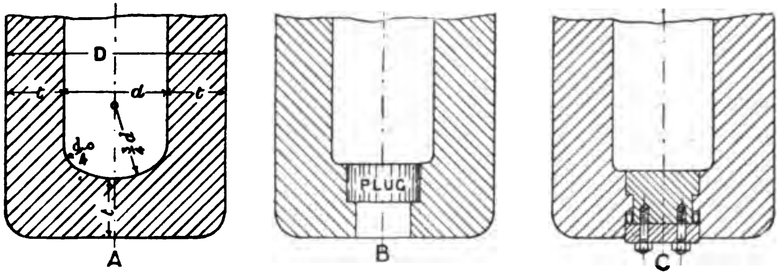


Fig. 359.

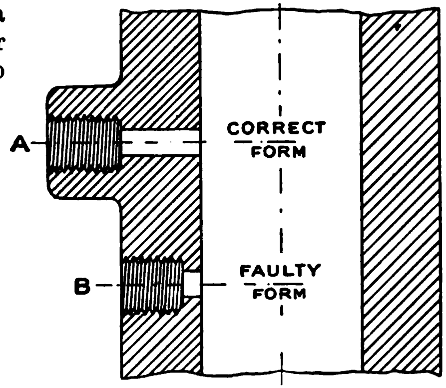


Fig. 360.—Use of strengthening boss.

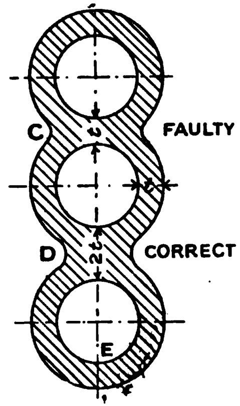


Fig. 361.—Thick partition.

in cylinder design, wherever a pipe connection is to be made to a cylinder, a *boss* A, Fig. 360, must be provided to make good the metal that has been cut away in drilling the hole, otherwise the cylinder will be materially weakened, as shown at B, Fig. 360.

When cylinders are cast together, the same want of skill is sometimes met with, the metal, *t*, between them being made the same thickness as the cylinder C, Fig. 361, instead of being twice that thickness (as it should be), as shown at D. For, obviously, when adjacent cylinders are simultaneously worked there is twice the circumferential tension at D that there is at E.

EXERCISES.

Designing, etc.

1. The length of a cast-iron steam pipe is 120′, and the working pressure 160 lbs. per sq. inch. How much will it expand in being heated from 32° F.?
2. A copper steam pipe is worked at 130 lbs. per sq. inch, and it has a length of 90″. How much will it expand in being heated from 32° F.?
3. Make a rough sketch design for a 6″ Armstrong hydraulic joint, pressure 800 lbs. per sq. inch; give the principal dimensions.

[1] When this occurs, there is always the possibility of the stress being increased in the internal layers, owing to the presence of the pressure water, and rupture of the press may happen at a pressure sensibly below the one it was designed for. A slight leakage will often take-up (or rust-up) after a few days by rusting of the pores.

[2] This is usually the bottom end of the cylinders.

Sketching Exercises.

4. Sketch in fairly good proportion three examples of cast-iron flanged joints for steam pipes.
5. Show by sketches three examples of how copper steam pipes are fitted with gun-metal flanges.
6. Show by sketches (*a*) a mild-steel solid drawn pipe with cast-steel flanges; (*b*) a welded iron pipe with riveted flanges.
7. Show by neat sketches the following joints used for tubing: ordinary socket, Perkin's, Perkin's joint with copper washer, and Emery's joint for connecting a steel tube to a gun-metal nozzle.
8. Sketch the following: boiler tube and ferrule, boiler stay-tube with backnuts, boiler tube with Admiralty ferrule. What is about the ordinary diameter and thickness of the last-mentioned tube?
9. Show by sketches the usual form of union joint. Under what conditions of working are these joints used?
10. Sketch three different expansion joints suitable for steam pipes, and say which you would prefer, and why.
11. Sketch an ordinary spigot and socket joint suitable for a cast-iron water pipe. Why is this type of joint most suitable for horizontal pipes?
12. Show by a sketch how you would form the bottom of a hydraulic cylinder so that there would be no danger of the bottom being forced out. In some cases it is necessary to cast the cylinder with a hole through the bottom; why is this? Show how such holes are afterwards plugged up.
13. Explain, with the assistance of sketches, some of the faults occasionally met with in the design of cylinder castings, particularly for hydraulic work.
14. Why is it necessary to cast hydraulic cylinders with *casting heads*? Make sketches of the usual forms of heads, and say which you prefer, and why.

Drawing Exercises.

15. Make working drawings of a spigot and socket joint, made with lead, for a 6″ pipe. Fig. 344.
16. Make sectional elevation and end view of a turned and bored spigot and socket joint for an 8″ cast-iron pipe. Fig. 346.
17. Make a sectional elevation, and a sectional end elevation, of an Armstrong hydraulic pipe joint for a pressure of 700 lbs. per sq. inch, and diameter 4″. You should show the detail of the *joint* full size. The other views half full size. Note.—The information given in Art. 148 will enable you to easily determine the leading dimensions.
18. Make a sectional elevation, plan, and sectional end elevation of a gland and stuffing-box expansion joint for a 6″ steam pipe. Scale half full size. (Refer to Fig. 441.)

CHAPTER XV

COTTERS AND COTTERED JOINTS

155. WHEN two rods are to be rigidly connected one to the other in such a way as to transmit a force in the direction of their length only, the most convenient joint for the purpose is the *Cottered or Keyed joint*, one form of which is shown in Figs. 362 and 363, where the end of the rod B takes the form of a *socket* or box, into which the end of the rod A fits and is held in position, by the *Cotter* C, a taper or wedge-like flat bar, which is driven through the socket and rod. The joint, as arranged in the figures, is a type suitable for connecting round bars to form a long pump rod for a well or mine, or some such purpose; but we shall see directly how it can be varied and adapted to the requirements of a number of interesting cases, and of these the best known is the joint which is used to connect the piston rod to the cross head of an engine, Fig. 367.

It can be shown[1] that when the joint (Figs. 362 and 363) is in tension its various parts will have practically the same strength when their proportions are as follows:—

PROPORTIONS OF COTTERED JOINTS FOR UNIFORM STRENGTH.

$d_2 = 1{\cdot}21d$	$d = 0{\cdot}82d_2$	$D_2 = 1{\cdot}75d$
$b = 1{\cdot}31d$	$t = \dfrac{d_2}{4}$	$D = 2{\cdot}42d$
$l = l_2 =$ from $0{\cdot}75d$ to d	$D_3 = 1{\cdot}4d$	$t_2 = 0{\cdot}42d$

156. Clearance of Cotters.—An important feature of the cottered joint is the clearance. Obviously, if the cotter in Fig. 362 is to draw the rod end A into the socket of B, there must be some clearance at m and p in the socket to allow the cotter in being driven to enter further into the socket, and there must be the same amount of clearance in the rod at K. In this type of joint the cotter when fitted is driven home, and the clearance is usually from $\frac{1}{16}''$ to $\frac{1}{8}''$.

157. Taper of Cotters.—Now, it is not difficult to prove that with surfaces slightly greasy *the total taper*[2] *must not exceed* 9°, if the cotter is not to slip back after being driven into the joint. This corresponds to a total taper or *draught*[3] of 1 in 7. But of

[1] For proofs refer to author's "Machine Design, etc.," p. 225.
[2] Unwin's "Machine Design," vol. i. p. 219.
[3] This draught is the trigonometrical tangent of the angle of taper, when one side of the cotter is square with the bar, as in Fig. 379, or twice the tangent of half the angle when the cotter is tapered both sides, as in Fig. 380.

course, for safety's sake, the taper is made a good deal less than this, usually about $\frac{1}{4}''$ to the foot to $\frac{1}{2}''$ to the foot. But when any special arrangement for keeping the cotter from slacking is made, its taper may be as large as the 1 in 7.

COTTERED JOINT FOR TENSION AND COMPRESSION RODS.

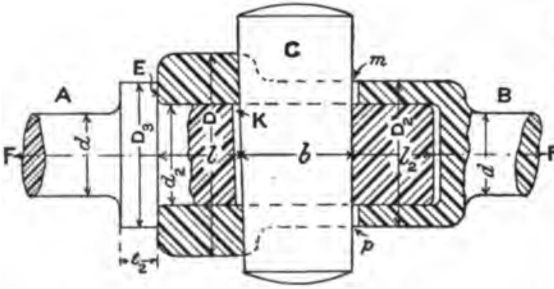


FIG. 362.

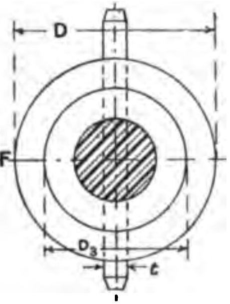


FIG. 363.

158. Proportions and Strength of Cottered Joints.—The young designer has, in these joints, good opportunities of proportioning the parts from suitable data; indeed, a little time could be most profitably spent in that way,[1] although in actual practice the empirical proportions we shall directly refer to are largely made use of, but even these, from an examination of several examples from ordinary good practice, seem to vary more than they should.

Of course, in calculating the dimensions in a given joint the nearest $\frac{1}{16}''$ is always used.

159. Various Cottered Joints.—Figs. 364 to 382 show representative examples of cottered joints, with the proportions in general use. Figs. 364 and 365 show two views of the bottom end of a wrought-iron standard cottered to a cast-iron bed plate, the unit being d, the diameter of the rod, in each case. Fig. 366 shows a low-pressure piston cottered to the rod of a tandem engine, d (the unit) being the mean diameter of the taper part. In Fig. 367 the socket of a cross head is thickened to give the requisite *bearing* surface for the cotter,[2] and Fig. 368 shows another arrangement of a cottered standard. Fig. 369 shows a cottered joint arranged for thrust and tension, the proportions being suitable for wrought-iron rods and steel cotter. A modified form of this joint is shown in Fig. 370, the socket being reduced in diameter below the part where the cotter bears upon it. The proportions are for all parts of wrought iron or steel. A simple cottered bolt is shown in Fig. 371 with the usual proportions. In Figs. 372, 373, and 374 we have two arrangements of foundation bolts and cast-iron washers;[3] in each case the bolts may be enlarged at the upper ends for the screwed part to a diameter $d' = \dfrac{d + 0{\cdot}05''}{0{\cdot}9}$. For other examples of foundation bolts, see Figs. 288 to 290 (Art. 127). The Figs. 375 and 376 show a cottered joint used for rough bars, the unit being the side S of the square bars. Fig. 377 shows a stud or bolt cottered in a casting, the hole usually being a cored one.

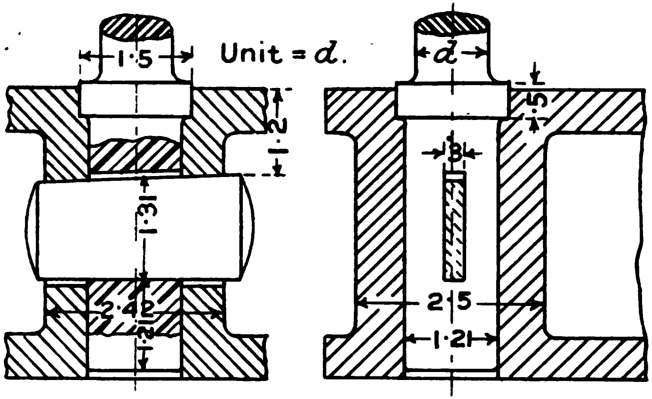
160. Use of a Gib.—Fig. 378 shows what would happen if a cotter AD was driven to draw the strap CB on to the rod EF. The friction between the cotter and strap at H would cause the latter to be sprung away, as shown dotted at B. To prevent this a gib G, Fig. 379, is used, but the hole in this case must be parallel (unlike the one in Fig. 378, where it has the same taper as the cotter)

[1] See the author's "Machine Design, etc." (p. 225), for an examination of the strength to resist various forms of rupture.

[2] The end of the rod is sometimes tapered (as in Fig. 366) both for piston rods and valve spindles, and, to facilitate the withdrawal of the rod, a small transverse hole through the socket may be drilled so that a taper pin can be used to wedge out the rod.

[3] When the cottered ends are round, as in these cases, it is usual to cast a snug or projection on the washer to prevent rotation of the cotter and bolt in screwing up.

VARIOUS COTTERED BOLTS AND JOINTS, ETC.



FIGS. 364 and 365.—W.I. standard and C.I. bed plate.

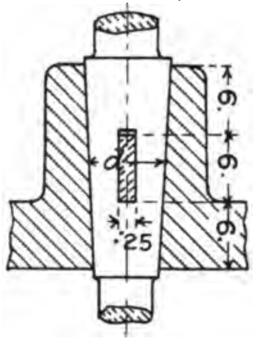


FIG. 366.—Steel rod and cotter. C.I. piston.

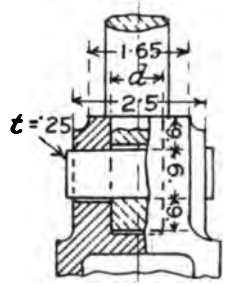


FIG. 367.—Piston rod and cross head.

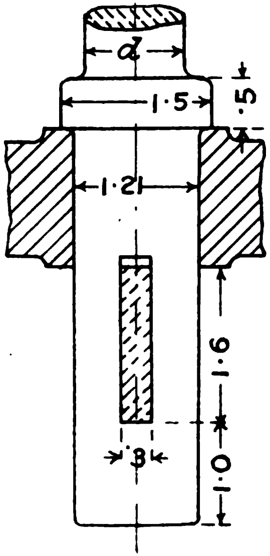


FIG. 368.—Rod and cotter either both W.I. or steel.

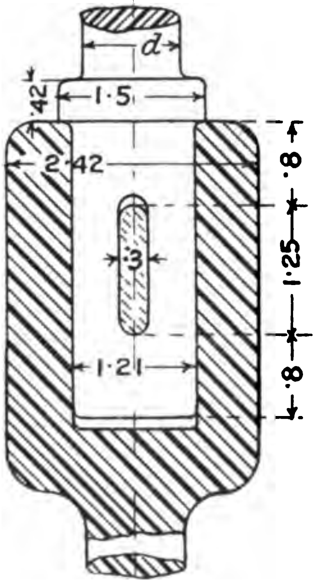


FIG. 369.—W.I. rods, steel cotter.

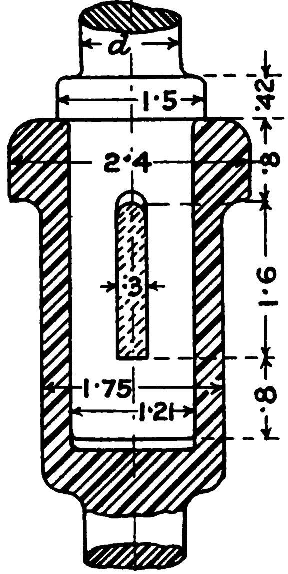


FIG. 370.—W.I. rods, steel cotter.

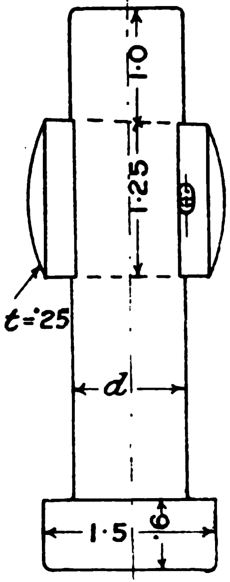


FIG. 371.—Cottered bolt.

and the taper given to the gib as at MN. Two gibs are sometimes used, as in Fig. 380; the taper is then either equally divided between them as shown, or only one gib need be tapered.

FURTHER EXAMPLES OF THE USE OF COTTERS.

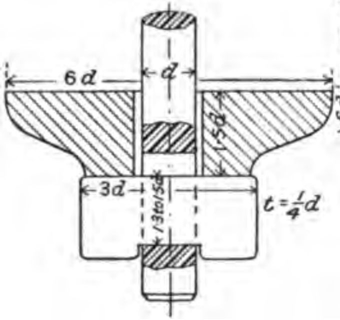
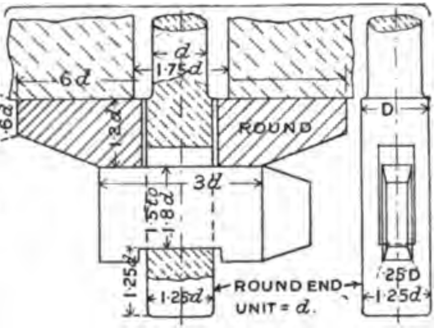
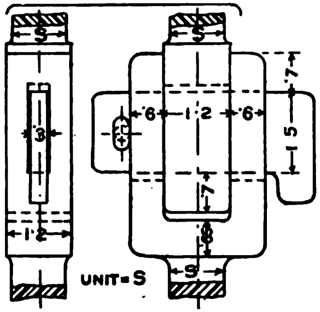


Fig. 372.—Foundation bolt. Rod and cotter both either W.I. or steel.



Figs. 373, 374.—Foundation bolt. Rod and cotter both either W.I. or steel.



Figs. 375, 376.—Cottered joint for rough bars.

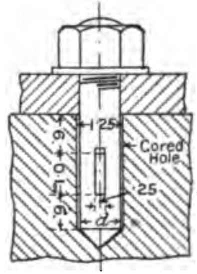


Fig. 377.—Bolt cottered into casting.

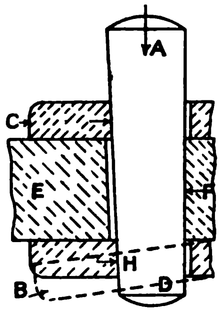


Fig. 378.—Cotter without gib.

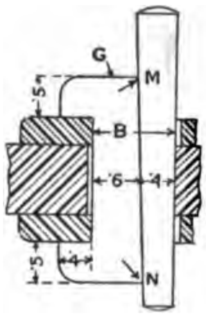


Fig. 379.—Use of gib.

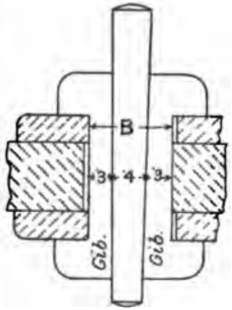
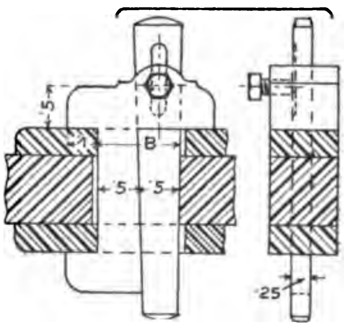


Fig. 380.—Cotter with double gibs.



Figs. 381, 382.—Gib and cotter with set-screw.

EXERCISES.

Drawing Exercises.

1. Make working drawings of a cottered joint for a 3″ round steel bar with steel cotter. It is to be suitable for a tensional and compressional condition of the bar, and to be of the form shown in Fig. 370. Scale full size.

2. Draw three views of a $1\frac{1}{4}$″ foundation bolt, with bottom end cottered and fitted with cast-iron washer (Figs. 373 and 374), both ends to be enlarged so that they are equal in strength to the body of the bolt. Scale full size.

3. Make suitable views of a cottered joint for rough 2″ square bars. Scale full size (Figs. 375 and 376).

4. The upper part of a machine is supported by four 3″ wrought-iron round standard, cottered into a cast-iron bed plate. Set out a suitable joint showing two sectional views (Figs. 364 and 365).

Sketching Exercises.

5. Make a freehand sketch of a cottered joint suitable for connecting lengths of a long pump rod.
6. Show by a sketch how a piston rod can be fixed to a piston by a cotter.
7. Make a sketch of a piston rod cottered to a cross head.
8. Show by a sketch any application of a cotter with double gibs (Fig. 380).
9. Sketch a bolt cottered into a casting. Under what circumstances would you use such a bolt?

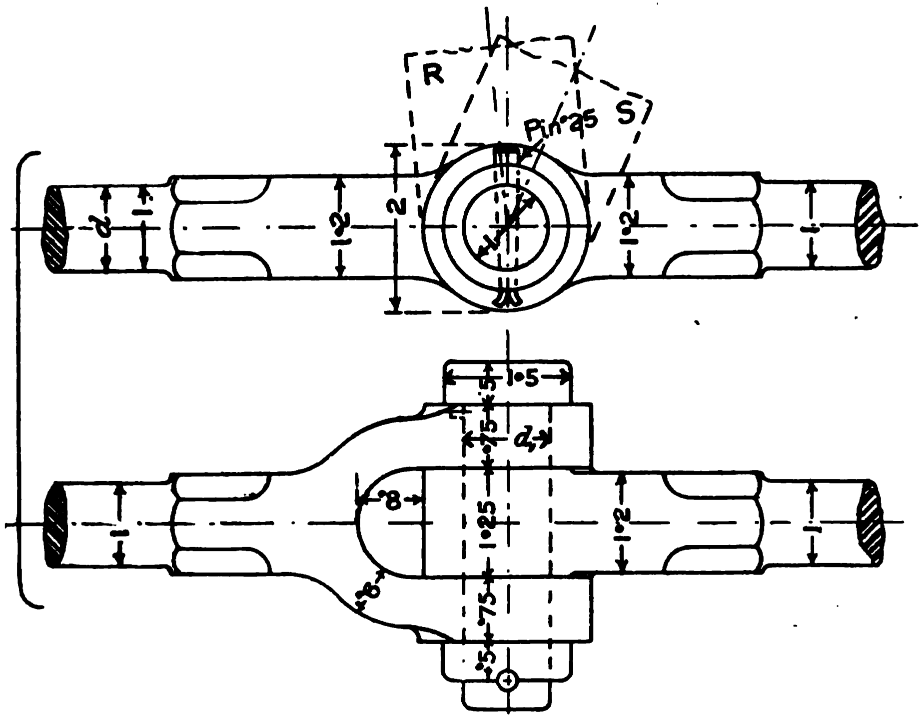
CHAPTER XVI

PIN OR KNUCKLE JOINTS, PITCH CHAINS, ETC.

161. PIN or knuckle joints are used (*a*) in structures, the pin connecting two or more bars or rods (such as the members of a braced girder, suspension chain, or roof principal) whose axes intersect in the axis of the pin, a special form being the forked knuckle joint, and another important form the eye joint of suspension links; (*b*) in machinery, to connect two rods or parts so that one may have a small angular movement about the other, the best known case of this kind being the joint connecting a valve rod to the rod of an eccentric; (*c*) for the joints of gearing and elevator chains.[1]

The following is an example of (*a*):

162. Forked Knuckle Joint.—Two views of this joint are shown in Figs. 383 and 384, and the dotted parts show how two other members R and S (usually split ones) are also sometimes connected by the joint, the opening of the fork being increased to accommodate them. The proportions of the joint in terms of *d*, the diameter of the rod, may be as given on the figures. No part of the joint will then be weaker than the rod when it is either in tension or compression; indeed, the pin could in some cases be made somewhat smaller, as we shall see in the next article, but making it the same size as the rod provides a margin of strength to resist the bending action which occurs when the pin is a somewhat loose fit or becomes worn.



FIGS. 383, 384.—Forked pin or knuckle joint.

163. Strength of Knuckle Joint Pin.—If there

[1] The pin joint of an engine cross head is also a special joint of this kind, but it is convenient to treat this separately.

is no bending[1] action, the pin will fail in double shear, and then, if d = diameter of the rod, d_1 = diameter of the pin, f_t = the tensional strength of rod per sq. inch, f_s = the shear strength of the pin per sq. inch, = say $0{\cdot}8f_t$,

Then $$d^2\frac{\pi}{4}f_t = 2\left(d_1^2\frac{\pi}{4}f_s\right), \text{ or } d_1 = 0{\cdot}79d \quad \ldots \quad (33\text{A})$$

164. Suspension Links, or *plate-link* chains are used in structures of the suspension bridge type, but the same type of chain only with short links and suitable pins has for years been increasingly used as **gearing chains**, when a positive and powerful drive is required. Figs. 386 and 387 show the forms and proportions of eyes (in terms of W) found by experiment to be strongest by Sir

FORMS OF SUSPENSION LINK EYES.

UNIT = W

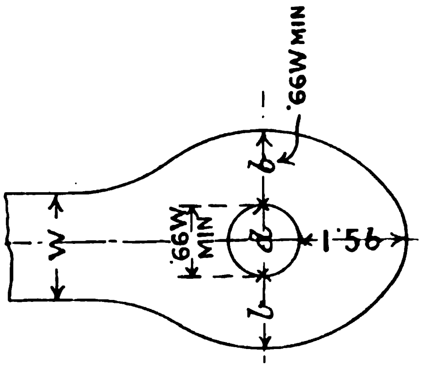


FIG. 385.—Hammered eye, American form.

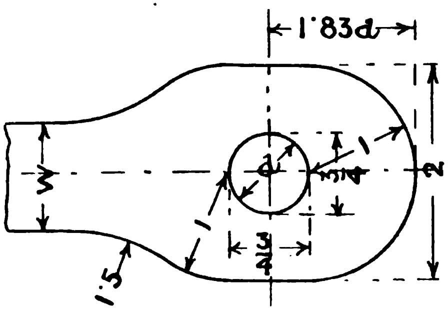


FIG. 386.—Berkley's form.

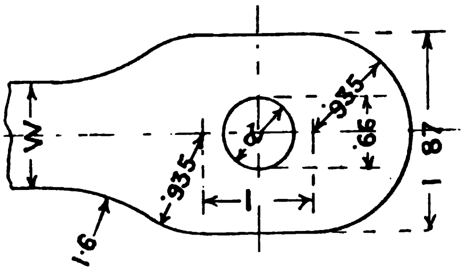


FIG. 387.—Fox's form.

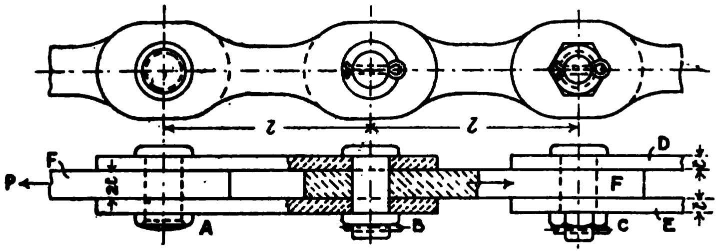
G. Berkley and Sir C. Fox respectively. It was found that if d is less than 0·66W the link crushes in the eye. In Fig. 385 is shown the **hammered eye** largely used in America. Figs. 388 and 389 show a chain arranged with two thin links, D and E, and a thick one F (double the thickness of the others) between them. Of course, the length l depends upon the flexibility required; it is sometimes as large as 25′. These figures also show the three different methods of fastening the pins. **A multiple link chain** is shown in Fig. 390. Of course, in this case, for uniform strength, the total thickness at M must equal the total thickness at N, or $3t = 4t_1$.

165. Gearing Chains.—The development of the motor-car and the popularity of the *chain-drive* have given a great impetus to the production of efficient pitch chains suitable to run at high speeds. For many years engineers have used chains for various special purposes in the transmission of power, but the possibilities of this system of transmission for many purposes,

[1] Unwin shows that where bending occurs and the joint is subjected to stresses reversing in sign, a skin stress of 5400 due to bending will not be exceeded if the diameter of the pin = $0{\cdot}0224\sqrt{P}$, where P = axial force in the rod.

when a *positive* drive with no slip is required, and where ordinary gearing would be inconvenient, have only been in recent years grasped. It frequently happens that two shafts can be advantageously connected by a pitch chain, where their distance apart

ARRANGEMENTS OF LINKS AND PIN FASTENINGS.

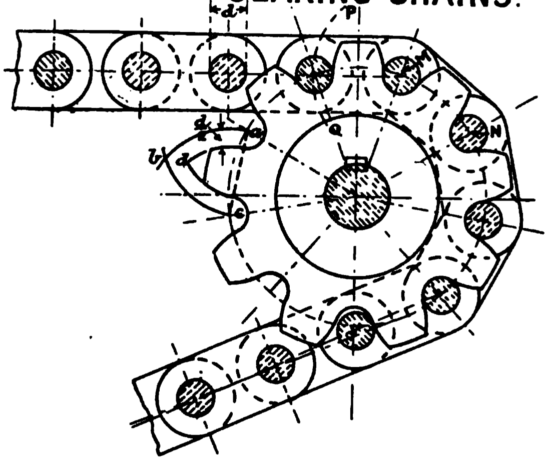


Riveted. Split pin. Nut and split pin.
FIGS. 388, 389.

MULTIPLE LINK CHAIN.

FIG. 390.

GEARING CHAINS.



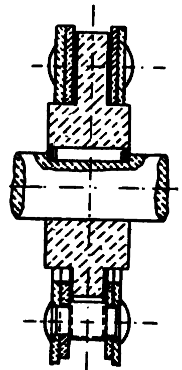
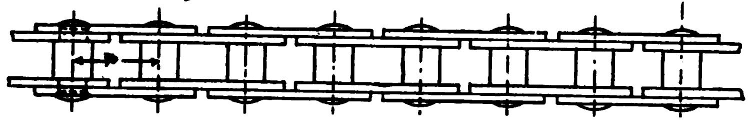


FIG. 393.



FIGS. 391, 392.—Single flat-link gearing chain.

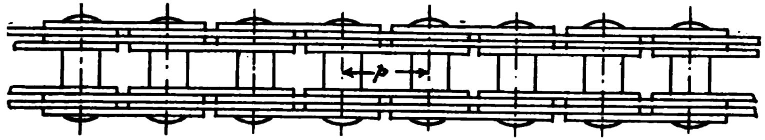


FIG. 394.—Double flat-link gearing chain.

is too small or their speed too slow to get an efficient transmission by belt. Thus, this gear or drive takes a sort of mid position between belting and ordinary gearing. The **primary fault** is one that rather seriously affects the durability of such drives, as no matter how well fitted the ordinary chains are to the teeth of the sprocket wheels on which they run, sooner or later the pitch of the chain becomes greater than that of the teeth on the wheels, causing them to work badly. However, as we shall see directly, efforts are being made to overcome this objection. Obviously, to reduce the trouble to a minimum the links should be made as short as practicable. The simplest form of pitch chain is shown in Figs. 391 to 393, a similar one with double links being shown in Fig. 394, but neither of these is fitted with rollers, as they are where the highest efficiency is required.

166. Form of the Wheel Teeth for Chains.—In Fig. 391 we have shown a sprocket or chain wheel. Now, if we had a perfectly *flexible* chain, the path of any point in the chain (such as the centre of a pin) as it left the wheel would be an *involute*[1] of a circle, and the actual curve of the tooth would be formed by drawing a parallel to the involute, distant from it half the diameter of a pin. But the links being solid, in most cases one pin P of a link (as the latter leaves the wheel) moves in an arc PQ about the centre M of its other pin. So, to set out the teeth, first find the centres *ac*, MN, etc., of the pin positions round the wheel, which form the corners of a polygon whose sides equal in length the pitch of the chain, the number of sides of course being equal to the number of teeth. Then, with centres *a* and *c* and radius *ac*, describe arcs intersecting in *b* (these are the paths of the pin centres), and with the same centres, radius equal to $ac - \frac{d}{2}$, describe arcs intersecting in *d*, which give the sides of the teeth, and the teeth can be completed, as shown, by giving them a suitable length.

EXERCISES.

Design and Drawing Exercises.

1. Make working drawings of the knuckle joint (Figs. 383 and 384) for a 2″ rod.
2. Make a working drawing of the end of a suspension link to take a load of twenty tons; the width of the link is 8″. You may make use of the proportions recommended by Berkley, and use a working tensile stress of five tons per square inch. What shear stress is the pin subjected to, and what crushing stress?
3. Draw two views of a 15-teeth sprocket wheel for a single flat-link gearing chain, whose pins are $\frac{5}{16}$″ diameter, and whose links have a pitch of 1″. Figs. 391 and 393.

Sketching Exercises.

4. Sketch three forms of eyes for suspension links, and give their usual proportions in terms of the width of the links.
5. Sketch a forked pin or knuckle joint. Figs. 383 and 384.

[1] See the author's "Geometrical Drawing," p. 160.

CHAPTER XVII

BEARINGS, JOURNALS, HANGERS, ETC.

167. THE parts of a shaft, spindle, or rotating piece which are supported by the bearings are called **Journals.** The simplest form of bearing is a cylindrical hole in the frame of the machine, such as is shown in Fig. 395, which is often met with in rough crane work. We have in this case a *Solid Bearing*, in the sense that it is not split, but in one piece. When end movement of a shaft or spindle of a machine [1] is to be prevented by the bearing, the journal is usually fitted with **solid collars**, as shown in Fig. 397, but of course this necessitates making the bearing with a cap, as shown in Fig. 398. Solid cast-iron bearings, if of ample proportions and made of hard tough cast iron, are used in some classes of work with most satisfactory results, and with very little wear. And, should the wear become excessive, they can be restored and fitted with **gun-metal bushes**, but in this case it is not always easy to bore them true to the original centres, so this is an additional reason for bushing them, as in Fig. 396, although it adds to the first cost.

TYPES OF BEARINGS.

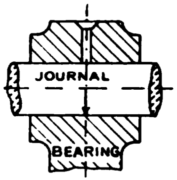


FIG. 395.—Solid bearing.

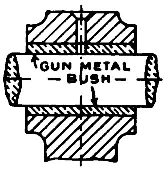


FIG. 396.—Bushed solid bearing.

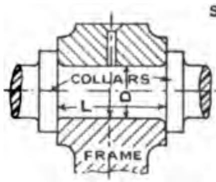


FIG. 397.—Journal with solid collars.

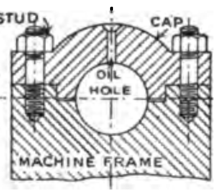


FIG. 398.—Bearing in two parts.

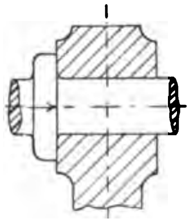


FIG. 399.—Collar bearing.

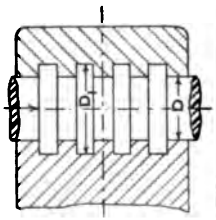


FIG. 400.—Thrust block.

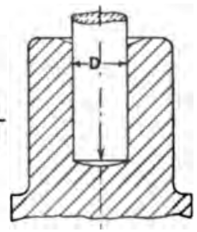


FIG. 401.—Footstep or toe bearing.

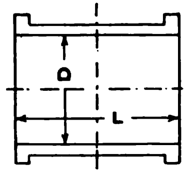


FIG. 402.—Area of bearing.

In each of the cases we have referred to the direction of the pressure on the bearing is perpendicular to the axis of the shaft. But two other typical cases occur when the main pressure is parallel to the axis. In the first, a **Thrust Bearing** either of the form Fig. 399, called **Collar Bearing**, or Fig. 400, with more than one collar, called a **Thrust Block**, is used. In the second, we have the case of the vertical shaft, where the end pressure is taken on what is called either a **Footstep, Toe, or Pivot Bearing.** In all of these cases the direction of the pressure on the bearing is indicated by an arrow in the figures.

[1] In the case of line shafting the end movement is prevented by loose collars, as shown in Figs. 86 and 558.

168. Effective Area of a Bearing.—The total load or pressure any bearing will support is the product of the *working pressure* allowable per sq. inch, and the *projected area,* the projection being taken in the direction of the load, on a plane at right angles to it. Thus, in the *Footstep Bearing,* Fig. 401, the projection is a circle,[1] and the area of the bearing surface will therefore be $D^2\frac{\pi}{4}$, and in the *Thrust Block,* Fig. 400, the area will be the sum of the areas of the collars, or $A = \frac{\pi}{4}(D_1^2 - D^2)N$, where N is the number of the collars. Then we have the important case of the *ordinary Horizontal Shaft,* Fig. 395, and Fig. 402 shows the projection of the bearing surface,[2] its area being $L \times D$, and this time the working pressure p equals the total load W, or $p = \frac{W}{L \times D}$.

VARIOUS BEARING ADJUSTMENTS.

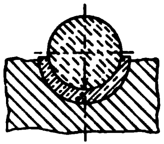


FIG. 403.

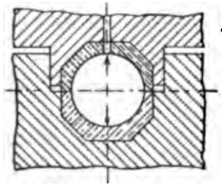


FIG. 404.

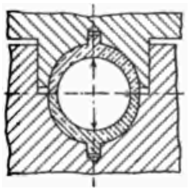


FIG. 405.

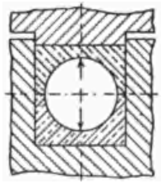


FIG. 406.

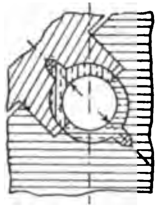


FIG. 407.

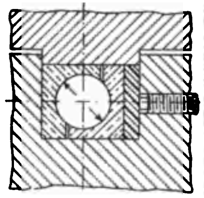


FIG. 408.

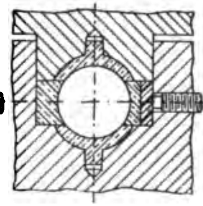


FIG. 409.

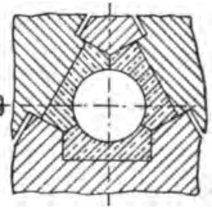


FIG. 410.

169. Various Bearing Adjustments.—In arranging bearings so that adjustments due to wear may be most effectively made, attention must be paid to the direction of the load on the bearing. Thus, in Fig. 403, the load is vertically downwards, and if it always acts in this direction a top *brass,* or *step* is not required; indeed, often the bearings for line shafting are fitted in this way, and with wood or shell caps to hold the lubricators and keep the dirt out. But in the crank shaft bearings of engines and similar machines the pressure acts alternately in opposite directions, and then two brasses are used, the dividing plane being perpendicular to the direction of the maximum pressure. Figs. 404 to 408 show five arrangements of this kind which speak for themselves, whilst

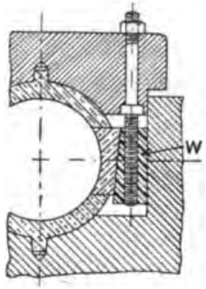


FIG. 411.

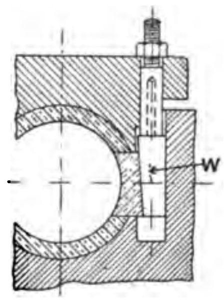


FIG. 412.

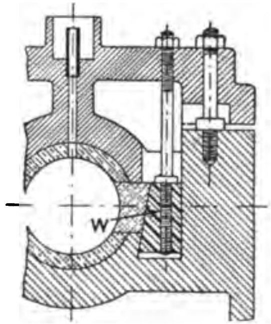


FIG. 413.

[1] Obviously this area is independent of any curvature that may be given to the end of the shaft.

[2] Of course, the real pressure between a journal and its bearing varies from point to point, and *p is a kind of mean value of the actual pressure.* Strangely enough, Mr. Box, in his well-known work on Mill Gearing, takes the area to be half the area of the journal, or $A = \frac{D\pi L}{2}$. This must be remembered should the student refer to Box's table of pressures.

in Figs. 409 to 413 five arrangements for dealing with more complex cases of varying pressure are shown. It will be noticed that in Fig. 412 the wedge end of the bolts for side adjustment W has its larger end at the top; the objection to this is that, should the nut work loose, the wedge is apt to work down and cause the packing piece to jamb the shaft. For this reason the arrangement shown in Figs. 411 and 413 is to be preferred. For the matter of that, there is much to be said against the practice which is so common, where large engines of the *stationary* type are concerned, of taking up the wear by means of wedges acting on the three or more brasses forming a bearing, for unless the greatest care is taken in the design, construction, and adjustment, no advantage will accrue from such refinements,[1] and the results are likely to compare unfavourably with the use of the simple ordinary two-part steps used by the locomotive and marine engineer.

170. Plummer Blocks or Pedestals.—The simplest form of Pedestal is the cast-iron **Bearing Block** shown in Figs. 414 and 415, which

BEARING BLOCK, OR SOLID PEDESTAL.

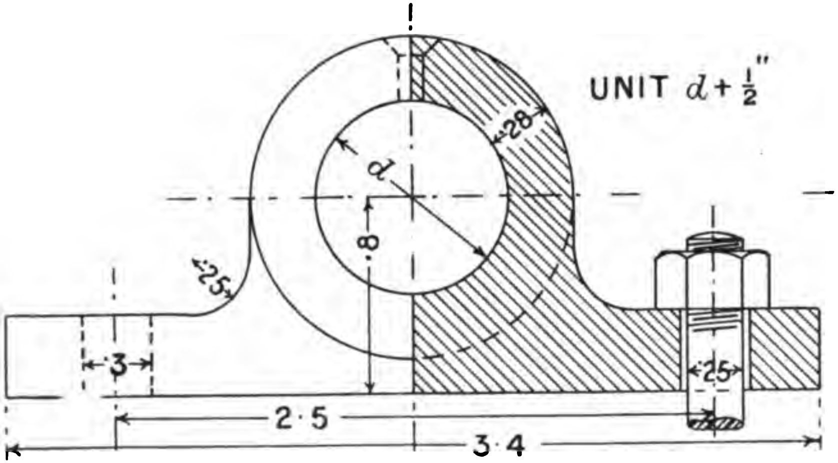


FIG. 414.—Sectional elevation.

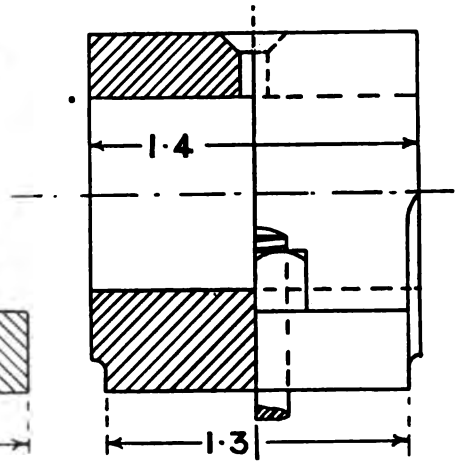
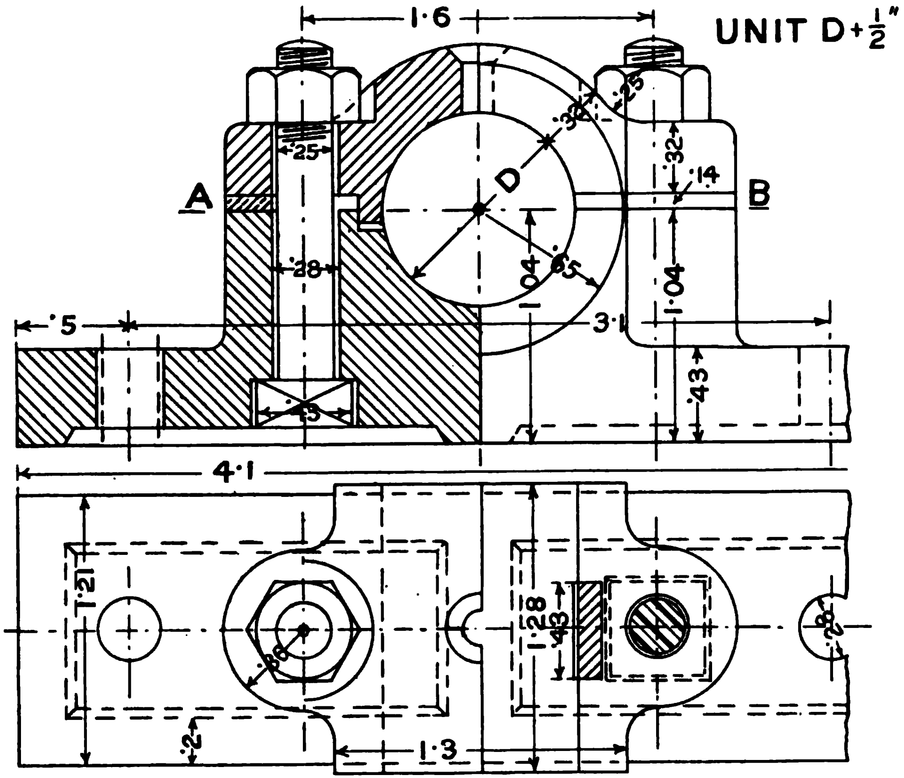


FIG. 415.—Sectional end elevation.

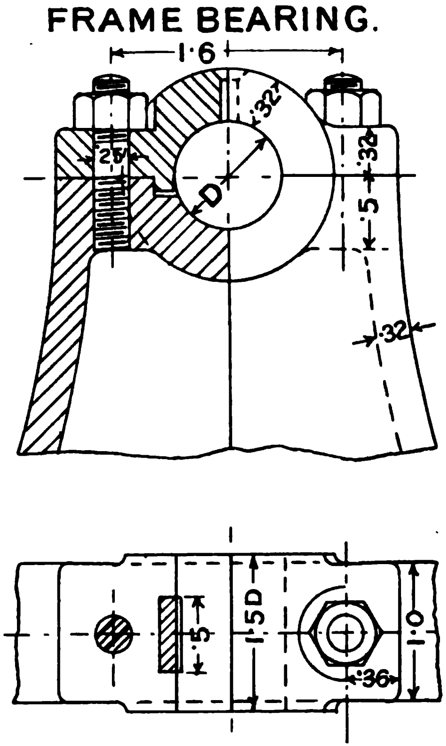
in this or some form varied to suit special jobs is largely used in some classes of rough work. Of course the oil hole is made in the part of the block which is highest when fixed. Suitable proportions are marked on it, the unit being $d + \frac{1}{2}''$. An improvement on this form is shown in Figs. 416 and 417, the bearing being fitted with a cap, so that, by filing the *packing piece*, or when none used, the top AB of the block, or the bottom of the cap, wear can be taken up. This form also allows of a shaft with collars being used. Figs. 418 and 419 show how this type of bearing is arranged to form part of the frame of a machine.

[1] Refer to Spooner's and Davey's "Elements of Machine Construction and Drawing," p. 49.

171. Ordinary Plummer Block or Pedestal, Unit = D + ½″.—An ordinary Plummer Block is shown in Figs. 420 and 421. The actual forms and proportions of these vary somewhat (for the same size shaftings) with different makers, but they all have the same essential parts, namely, the *block* B, *cap* C, *brasses* or *steps* S, and *bolts* E.



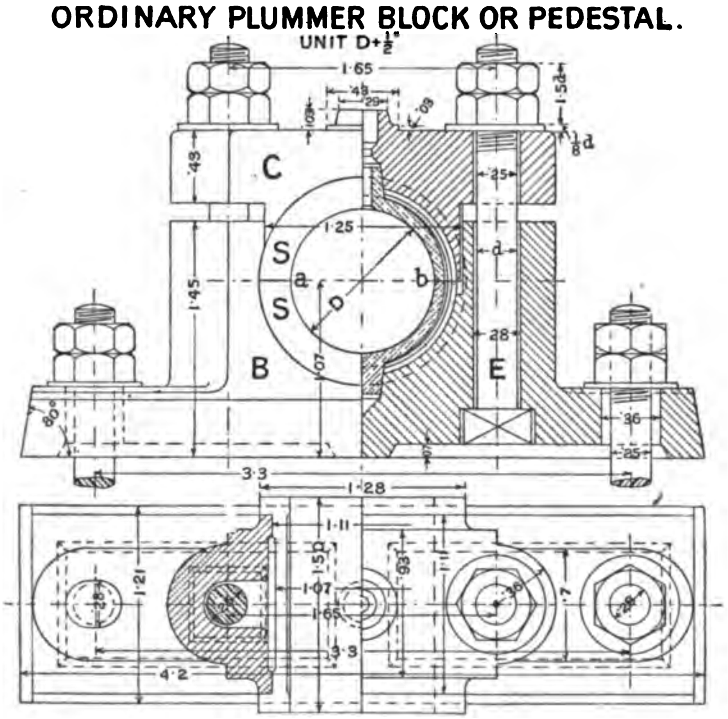
Figs. 416, 417.—Adjustable cast-iron bearing.



Figs. 418, 419.

The advantage of fitting the blocks with brasses of the shape shown is that they can have their fitting edges turned, and the block and cap bored to correspond. But for the heaviest work the old-fashioned brasses, Figs. 427G and 427H, with backs of octagonal form,

cannot be surpassed, as with them, when properly fitted, the whole of the bottom surface of the lower brass is supported by the block, and the flow of the brass under pressure and jarring is prevented. Of course the brasses are filed at the joints *a* and *b* (Fig. 420) to take up wear when necessary, but this has to be done with great care, or the shaft is held tight when the cap is bolted down.



FIGS. 420, 421.'

172. Seller's Self-adjusting Pedestal.—We have explained that cast-iron bearings run remarkably well when properly proportioned; indeed, with a length of four diameters and efficient lubrication, such bearings show little sign of wear after a lengthy run of the shaft at high speeds, so long as the pressure does not exceed 50 lbs. per sq. inch, and there is perfect alignment of the shaft. To secure this Mr. Seller designed his Pedestal, Figs. 422 and 423, in which it will be seen that the seats A and B of the cast-iron steps are spherical, and that so long as the pedestal has been fixed the right height and in the correct position, the steps can adjust themselves to a slight extent to the position of the shaft. The centre cup C is for ordinary lubrication, but the side cups T are fitted with a mixture of oil and tallow, which at ordinary temperatures is solid, but melts should the bearing be heated up to about 100° F., and in so doing protects the shaft from injury; the drippings falling into the side cups D.

173. 3" Plummer Block or Pedestal (Drawing Exercise).—The parts of a 3" pedestal are shown separated in Fig. 427, and the student should now be able to assemble them.

Instructions.—The following views of the complete block may be shown. (1) Elevation, (2) Plan, (3) Section on Line AB, (4) End Elevation. *Each view is to be properly projected and completed.* Then write upon each part the name of the material of which it should be made. Show chipping and facing pieces to block and brass, and lighten out brasses to save material and minimize labour; show long holes in block for adjustment, and how you would make the oil channels. **Scale, half full size.**

174. Crank Shaft Bearing (Drawing Exercise).—Parts of an adjustable bearing for the crank shaft of a horizontal engine are shown in Figs. 424, 425, and 426. The adjustment for horizontal wear is made by screwing up the wedge pieces, as explained in Art. 169.

' **Instructions.**—*Elementary students* may draw the bearing complete, placing the various parts in their proper positions, and showing the left-hand half of view Z in section on line AA and the right-hand half in elevation. Project from it an end elevation, as seen when looking in the direction of the arrow X. The plan need not be drawn, but dimensions for other views taken from it. **Scale, quarter full size.**

175. As a more **advanced exercise** *Students* may draw the bearing completely put together, showing—the left-hand half of view Z in section on line AA, and right-hand half in elevation, also a complete plan, and a vertical section on line MM, when looking in the direction of the arrow X. **Scale, quarter full size.**

SELLER'S SELF-ADJUSTING PEDESTAL.

WITH CAST IRON BEARING

UNIT D + ½"

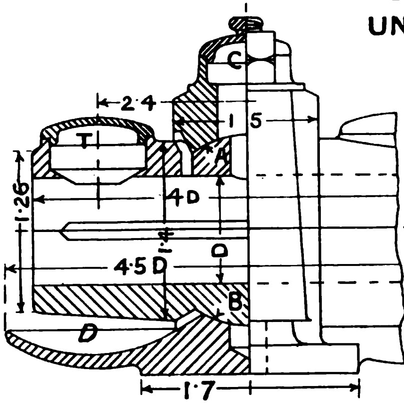


FIG. 422.

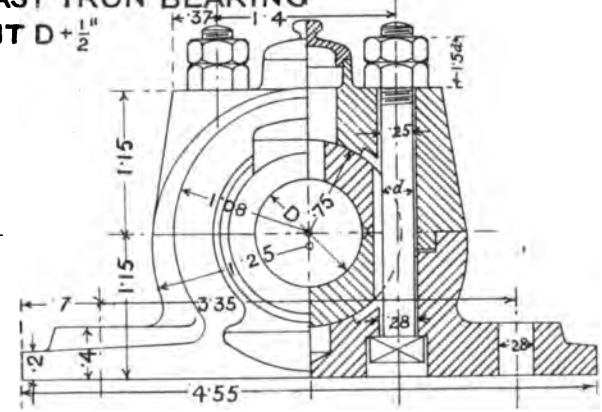


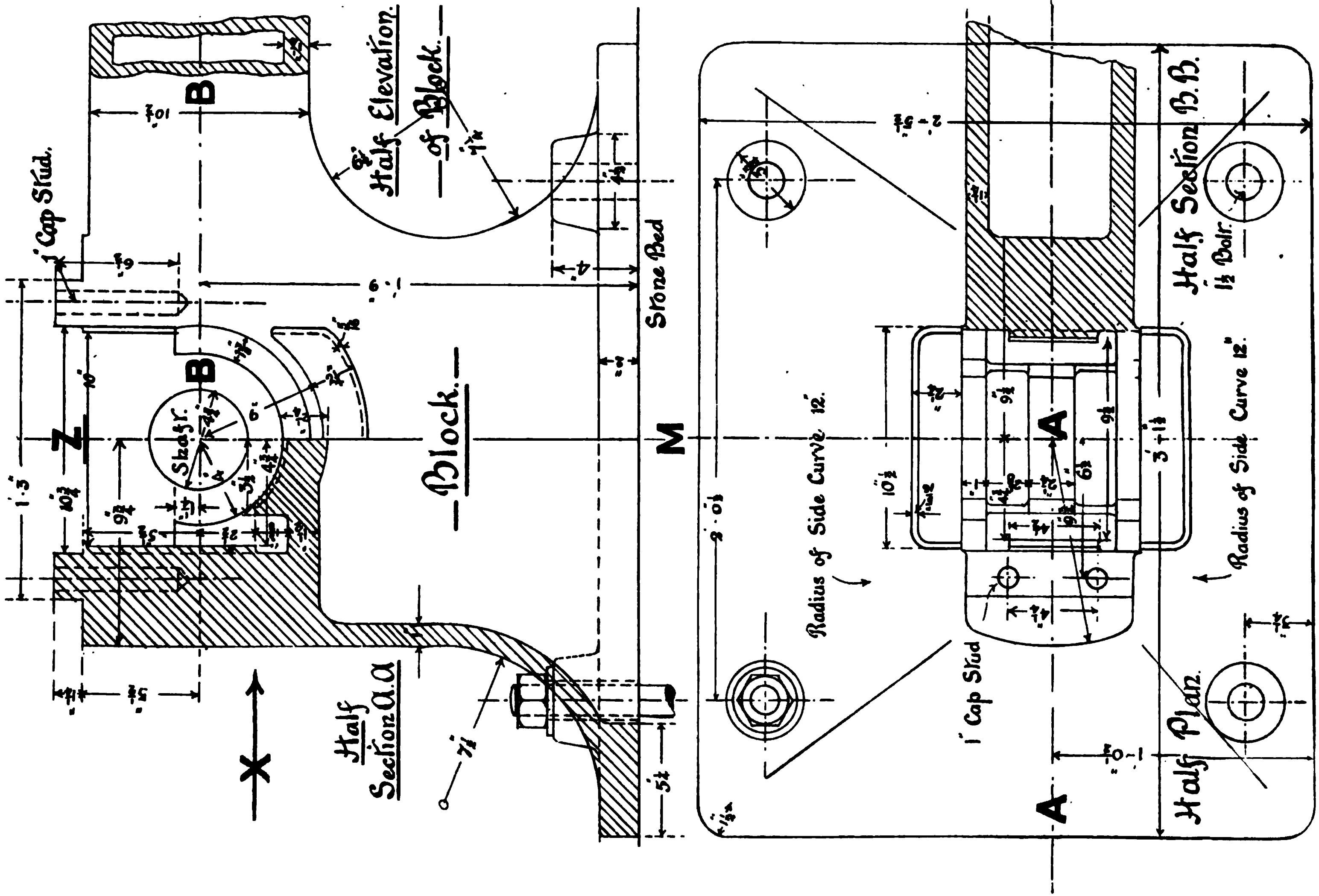
FIG. 423.

176. Brasses, or Steps.—We have seen, Art. 169, that certain bearings are fitted with brasses (or *steps*, as they are sometimes called); but the names do not really indicate the material, as they are usually made of **gun-metal** or an alloy of that type, such as **phosphor** or **manganese bronze**. There are several ways of forming them and fitting them to the supporting surfaces of the bearings, of which they form part, shown in Figs. 427A to 427L. The *unit* for the proportions is usually t, the thickness of that part of the brass which supports the load, and this may be for average thicknesses, $t = 0{\cdot}08D + 0{\cdot}1''$. The brasses shown in Figs. 427A, 427C, 427I, 427M, 427O, and 427S, are usually fitted by turning the fitting-strips (on backs) and boring the block or bed which receives them. Rotation of the brasses in the block and cup is prevented by either a stop-pin, as in Figs. 427A and 427B; by stop lugs, as in Figs. 427C and 427D; by rectangular chipping strips, as in Figs. 427K and 427L; and by (for Figs. 427I, 427M, 427O) the top of the upper brass being flat, the cap keeping them in position, or, should the back of the upper brass be also round, the *cap* is fitted with lugs which hold the packing pieces tight, and prevent rotation.

177. White Metal Bearings.—Many bearings are now fitted with white metal, or Babbit's anti-friction metal.[1] There are several ways of doing this. Fig. 427Q shows the metal run into the grooved bed of the bearing when the shaft is in position. In Figs. 427M, 427N, the metal has been run into the brass, caulked, and the hole bored in the usual way, whilst in Fig. 427R the spiral grooves, and in Figs. 427O, 427P, the round holes, are filled with the white metal.[2] Another method is shown in Fig. 427S, longitudinal strips of the white metal being *fitted* and driven into the grooves. Owing to the contraction of some of these alloys in

[1] An alloy of copper, tin, and antimony. The employment of these so-called anti-friction, soft, white metals is in the nature of a makeshift, and is largely due to the *heating troubles* which are met with when ordinary bronzes are used. Another very important **anti-friction metal is Perkins'**. It is an alloy of tin and copper in the proportion of 5 to 16, and is whitish in colour, but, unlike the white metals referred to, is *very hard* and exceedingly *brittle*, and the author's experience is that it makes admirable piston rings and slide-valve faces for very high steam pressures with or without lubrication, when exceptional care is taken to prevent fracture, the rubbing surfaces becoming very smooth and mirror-like.

[2] These expedients are employed to prevent flow of the soft white metal under pressure. This metal must always be encased by a metal such as bronze or cast iron, strong enough for the purpose to prevent such flow.



FIGS. 424, 425.—Bearing for crank shaft of horizontal engine.

DRAWING EXERCISE.

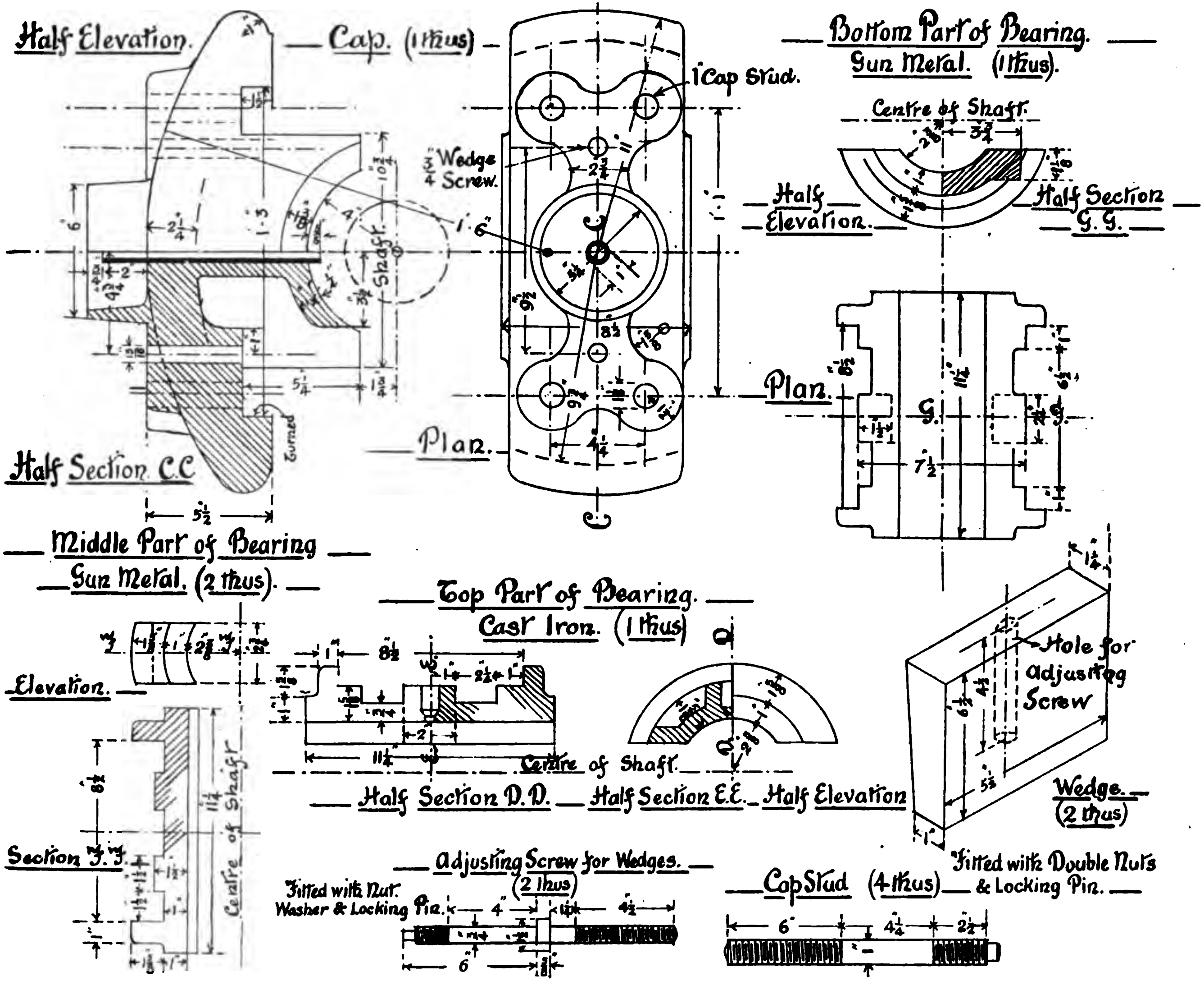


FIG. 426.—Details of bearing for crank shaft of horizontal engine.

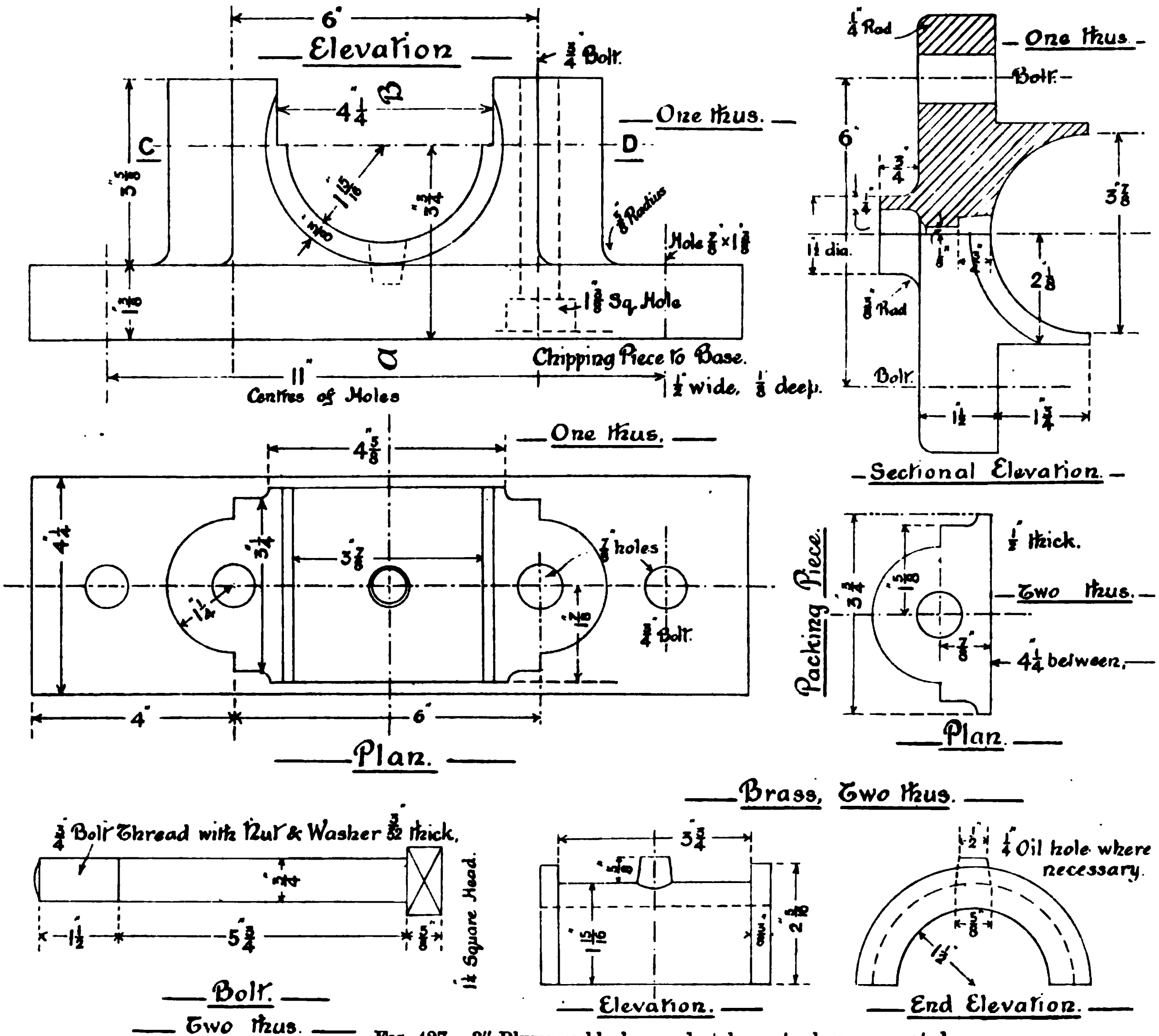


Fig. 427.—3″ Plummer block or pedestal: parts shown separated.

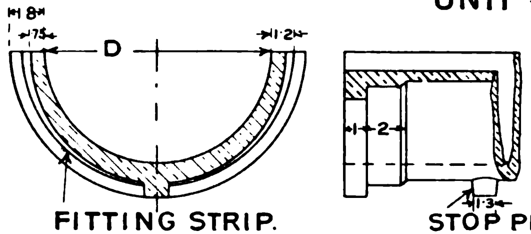
cooling, it is sometimes necessary in large bearings to somewhat expand the metal by hammering, to prevent the pieces working loose. This is best done by a few good blows on a piece of round lead of nearly the same curvature as the hole; as light blows on the surface of the white metal are apt to cause the latter to crumble away.

178. Hangers.—The arrangement of a bearing which supports shafting from the ceiling joists or beams is called a *hanger*. Figs. 427T, 427U, show one form, being another modification of the ingenious bearing designed by Seller, and shown in Figs. 422 and 423. The *unit* in each case = D + ½", and proportions are given as a rough guide, but, in most cases, for economical designs, the heaviness of the work must of course be considered and the scantlings modified with judgment.

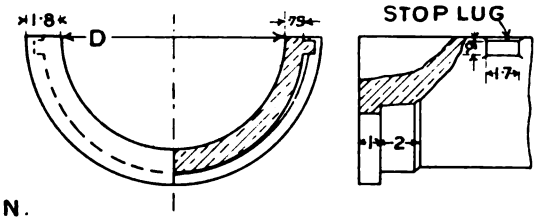
179. Wall Brackets are bolted to walls and used to support the Plummer blocks or pedestals. Two examples are shown: the first, Figs. 428 to 430, is for average work; while the second, Figs. 431 and 432, is for light work. Of course, A and B (of the first) must be made to suit the length and breadth of the base of the pedestal used. In this case one with a short base, collar holding down bolts being used for the cap. The difficulty in dealing with the bolt heads for the pedestal is overcome by putting the web WF out of the centre of the flange to make room for the bolt heads H. The proportional parts, in terms of the unit = D + ½", are shown.

VARIOUS BRASSES, OR STEPS.

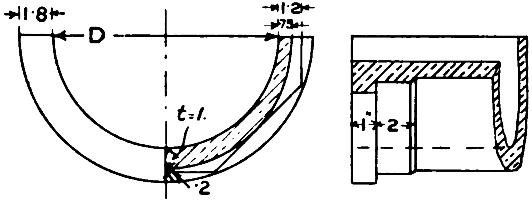
UNIT = ·08D + 0·1".



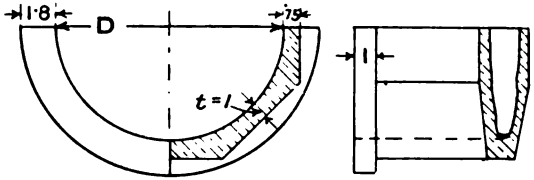
FIGS. 427a & 427b.



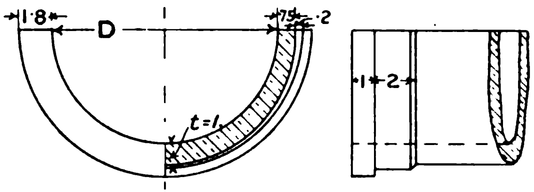
FIGS. 427c & 427d.



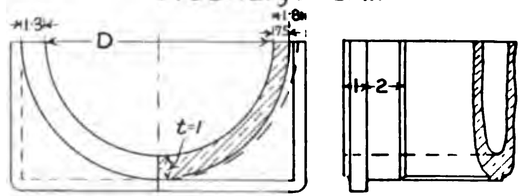
FIGS. 427e & 427f.



FIGS. 427g & 427h.



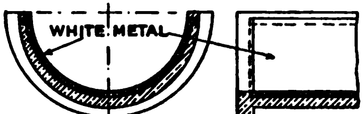
FIGS. 427i & 427j



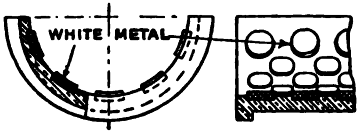
FIGS. 427k & 427l

180. Light Wall Bracket (Drawing Exercise).—Two views of a light wall bracket suitable for supporting a pedestal are shown in Figs. 431 and 432, and the following is a suitable exercise on it.

WHITE METAL BEARINGS.



Figs. 427M, 427N.



Figs. 427O, 427P.

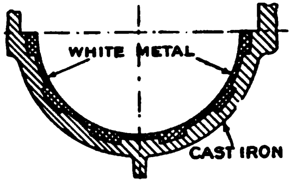


Fig. 427Q.

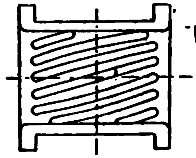


Fig. 427R.

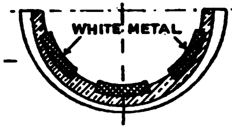


Fig. 427S.

ADJUSTABLE HANGER BEARING
SELLER'S TYPE

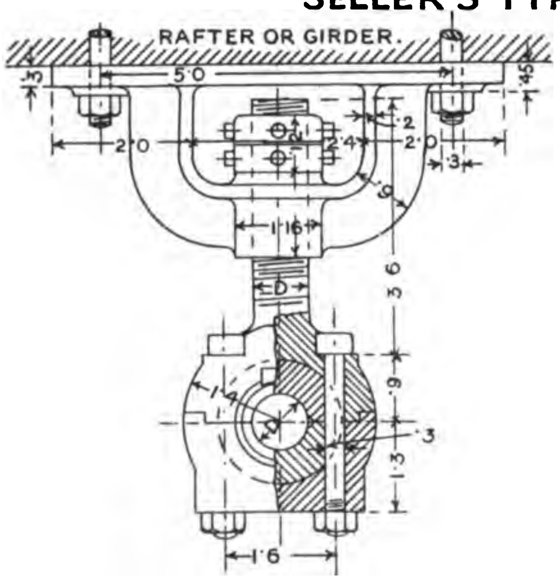


Fig. 427T.

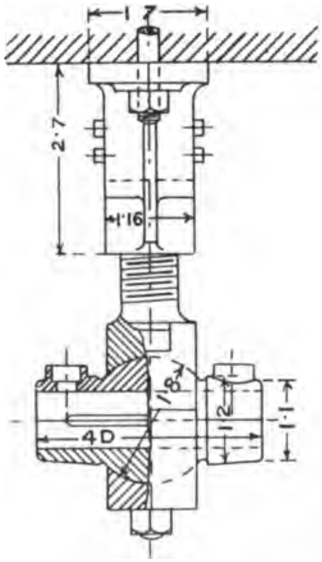


Fig. 427U.

Instructions.—Draw the side elevation and plan, also a front elevation as seen when looking in the direction of the arrow Z, and a section on line GH. **Scale, half full size.**

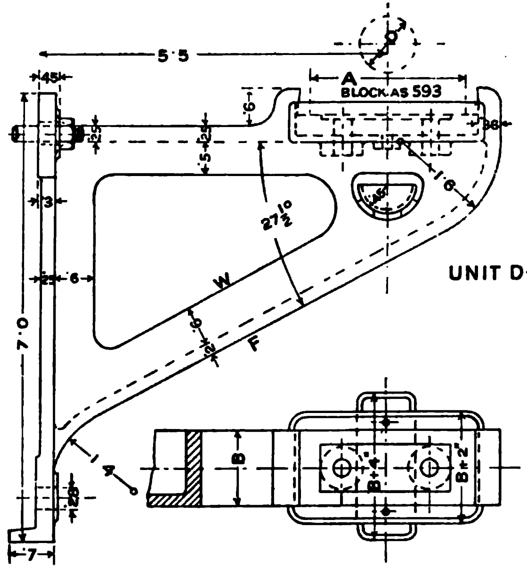
181. Wall Boxes for Plummer's Blocks or Pedestals.—In cases where a shaft is supported by a wall, a *wall-box* is used arranged so that when it is built into the wall the top of the box supports the wall above it, whilst the pedestal is bolted to the table of the box. Figs. 433, 434 show the ordinary form, for light work, and Figs. 435 and 436 one for heavier work, the arched top being a stronger form. T-headed bolts being used, the table or bridge can be arranged in a lower position or built up from the bottom plate, and the size of the box reduced. But in fixing such a box care must be taken to pack it all round, particularly at the top of each side, to prevent the thrust from the arched top spreading out the sides. As very large boxes cast in one piece are apt to crack, they are always made in separate plates bolted together, arranged in such a way that the brickwork is supported by abutting surfaces of the plates. When alterations necessitate the use of wall boxes, the latter are usually less strained by the

supported brickwork than they would be by new work which settles slightly, and the necessity to use built-up boxes is not so great. But in new work important boxes should be bedded on an ashlar stone with cement, another stone being placed over the box, resting on the brickwork at each side to form a lintel.

The points to receive attention in designing wall brackets and boxes are *accessibility* to all parts for adjustment and repairs, and for oiling and cleaning, and giving them such forms and proportions as to ensure proper stability and strength; and, when advisable, providing them with flanges along the inner edges for the attachment of wrought-iron **fire-proof plates**. Boxes are sometimes used merely to secure **wall openings** required for the passage of belts, shafts, ropes, etc. Such openings should be made as small as possible. Often for such purposes boxes of **circular form** are used, with a suitable flange.

182. Footstep, or Pivot Bearings.—The lower ends of vertical shafts are usually supported by *footstep* or *pivot bearings.* The ordinary form of this bearing is shown in Fig. 437, the end of the shaft being **steeled**, or it has welded to it a *steel end.* It is supported by a *steel disc*, slightly cup-shaped, whose rotation is prevented by the stop pin P; the gun-metal bush, which prevents lateral movements, is kept from rotating with the shaft by the snug S. The base plates of all these bearings, shown in Figs. 437 to 440, are square, and held down by four bolts, but of course in special cases they are designed to suit the sole plates to which they would be fixed. Fig. 438 shows another arrangement, the shaft being supported by a hard gun-metal disc block, with a hemispherical base, so as to rectify any slight movement of the block in fixing, or due to settlement. Rotation of the bush and pivot disc is prevented by the feather F, and the stop pin P, respectively. Three oil grooves meeting in the centre are cut in the disc. Fig. 439 shows an arrangement where four loose discs support the shaft, the oil being introduced at their centre, away from which it moves by centrifugal force, and in so doing lubricates the surfaces in rubbing contact. With this arrangement, should the lubrication between any two discs become momentarily faulty, rubbing occurs only between the other surfaces. The arrows show how the oil circulates. A cleaning hole is shown closed by the set-screw S.

WALL BRACKET FOR PEDESTAL (HEAVY TYPE).



FIGS. 428, 429.

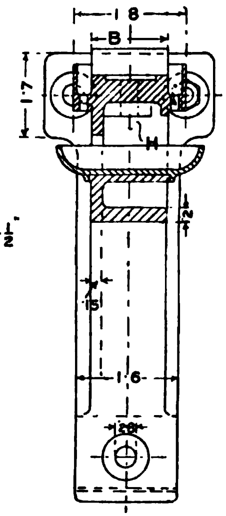
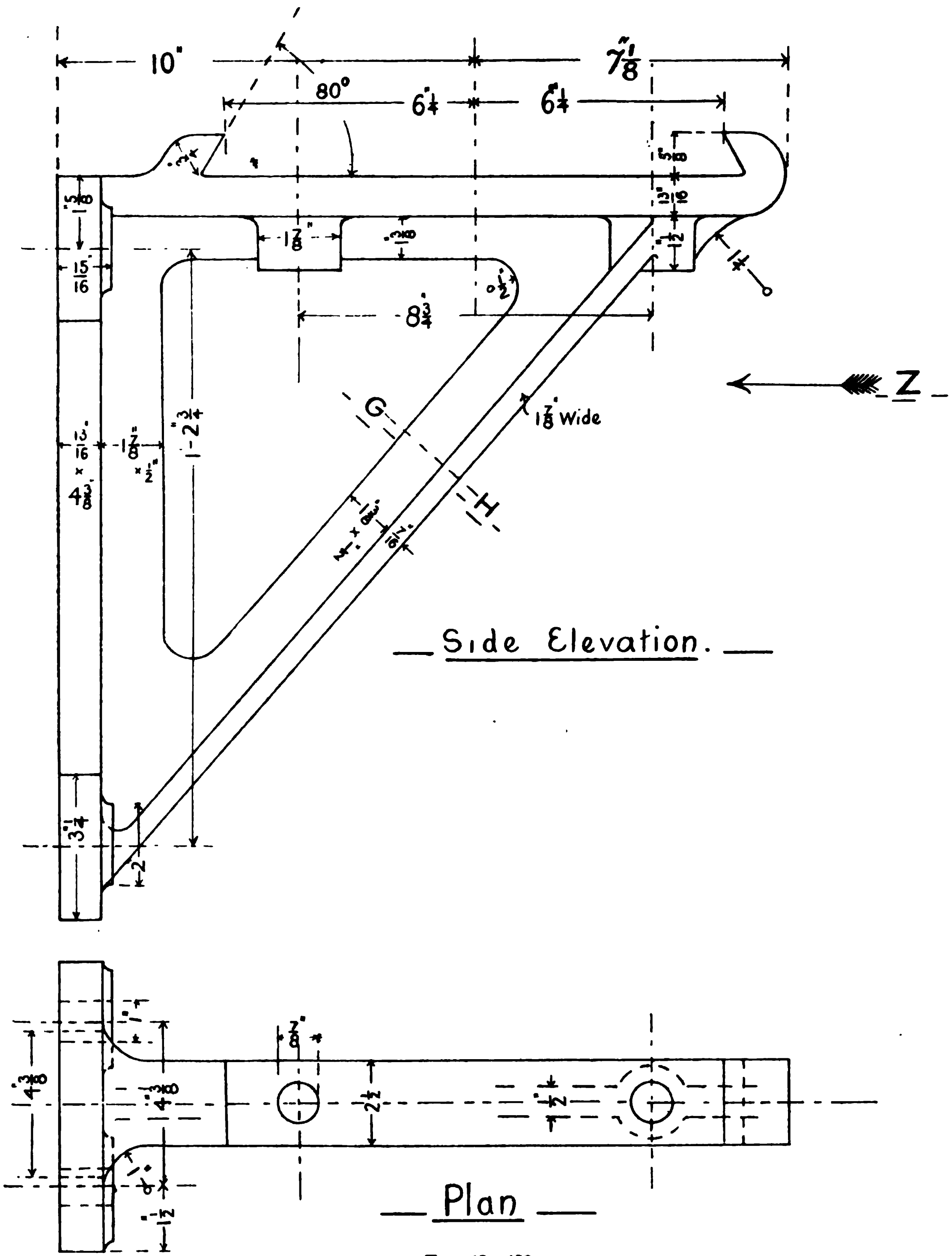


FIG. 430.

When great loads have to be supported by vertical shafts, it may happen that the pressure on the pivot would be too great for the bearing to work satisfactorily, but it is not desirable to load a footstep bearing to more than 200 lbs. per sq. inch, although pressures up to 400 lbs. are possible by carefully attending to lubrication.[1] Then **Collar Bearings**, one of which is shown in Fig. 440,

[1] With forced circulation of oil a pressure of 1 ton per sq. inch on the pivot of a 5" shaft has been found workable.

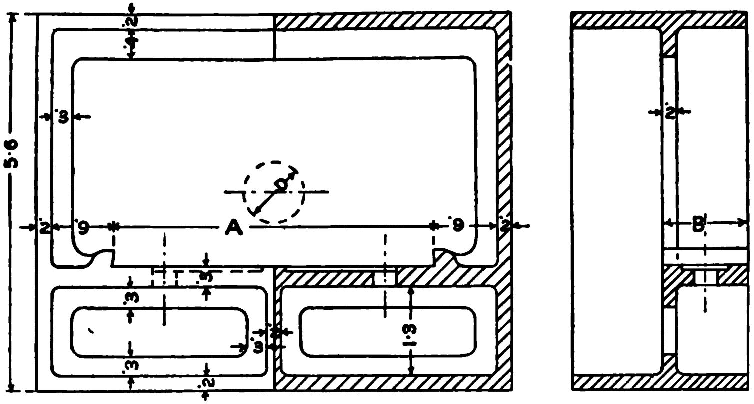
WALL BRACKET.



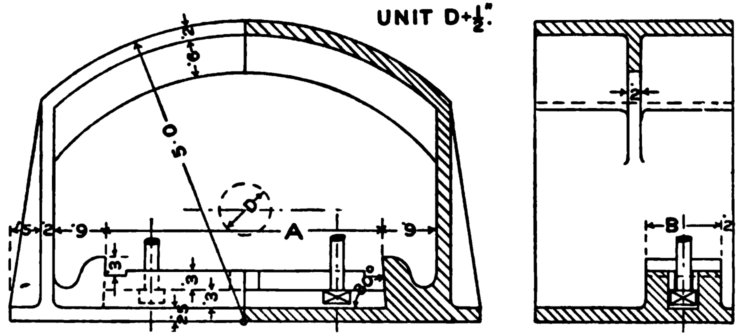
Figs. 431, 432.

are generally used.[1] Of course, with the arrangement shown, the bush must be split, the two parts being made to clip the shaft before it is lowered into the casting.

WALL BOXES FOR PEDESTALS.



FIGS. 433, 434.—Ordinary form for light work.



FIGS. 435, 436.—Arched-top type for heavy work.

FOOTSTEP, OR PIVOT BEARINGS.

UNIT = D

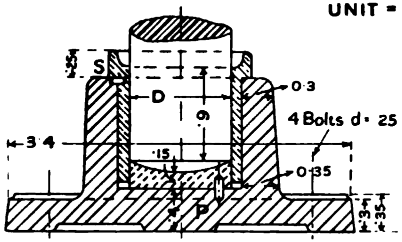


FIG. 437.—Ordinary form.

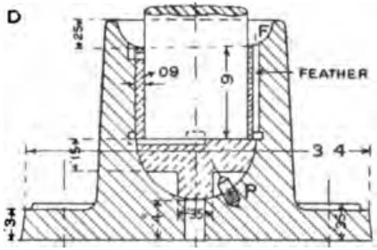


FIG. 438.—Footstep with hemispherical seat.

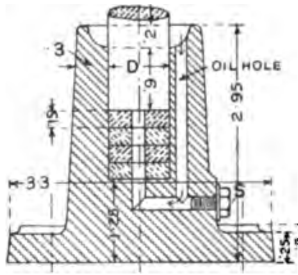


FIG. 439.—Footstep with multiple discs.

UNIT = D + ½"

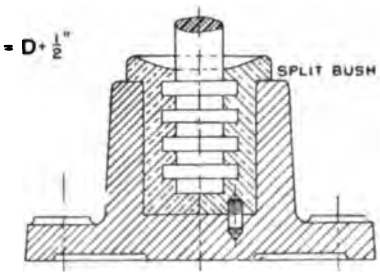
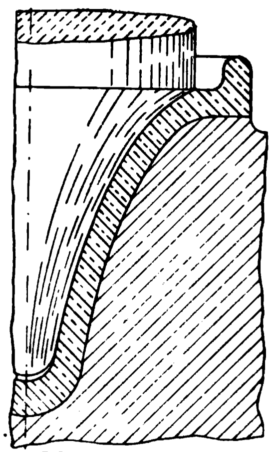


FIG. 440.—Footstep collar bearing.

183. Schiele's Pivot, Fig. 441, wears equally on all rings or diameters. It was designed to overcome the inequality of wear

[1] If r_1 and R be the inner and outer radii of the collars, then the moment of friction $M = \frac{2R^3 - r^3}{3R^2 - r^2} W\mu$, where W is the total load on the collars, and μ the coefficient of friction. The work done in inch-lbs. per minute $= MN2\pi$, where N = revolutions per minute. Refer to author's "Machine Design, etc.," p. 265.

referred to in the previous article. As the wear of the surface is uniform at every ring, the two parts always fit each other accurately, and the pressure is always uniformly distributed, and never becomes so intense at certain rings (as is the case in other pivots) as to force out the lubricant and grind the surfaces. The curve is a *tractrix*,[1] and the shape of the pivot is formed by revolving the curve about its axis. This pivot (sometimes called, strangely enough, *anti-friction*) is not often used, as its shape is not very convenient; it is expensive to manufacture, and, as compared with a **flat pivot**, it wastes **50 per cent. more energy in overcoming friction**, as is seen by comparing their respective moments of friction.[2] But the advantage of this pivot is in the fact that it **maintains its shape** as it wears, and is self-adjusting. Obviously, as the pressure p is constant, the radius $R \propto \cos a$, where a is the angle the normal to the curve makes with the axis. This pivot has its **maximum supporting area** when the constant length of the tangent intercepted between the two rectangular arcs bounding the curve equals R, the radius of the shaft; then **the moment of friction**, M, is $= W\mu R$, where W is the load, its value being the same as for a **spherical pivot**.



SCHIELE'S PIVOT.
FIG. 441.

184. Materials used for Bearings, etc.—We have seen, Art. 172, that *cast iron*[3] can be efficiently used for moderate pressures; this is largely due to its porosity and absorptive power, and the persistent way in which grease and oil adhere to it. The most suitable material for a bearing depends upon the material of the shaft journals, the pressure, speed and lubricant, and no very strict rule can very well be laid down. *Wrought iron*, if free from surface defects, is a good material for journals, and *mild steel*, which is more homogeneous, still better; whilst *hard steel*, ground to shape and well bedded in its bearing, will carry an enormous pressure if efficiently lubricated and its temperature is kept down.[4] It does not often happen that the bearing and journal are made of the same material, as it is generally advisable to make the bearing of a softer metal so that most of the wear takes place there, as its renewal is an easier and less expensive matter, and the strength of the journal is not reduced by sensible wear, as it would be if cut and worn by running in a harder bearing. For these reasons bearings are often **babbitted**, or lined with soft white alloys, as explained in Art. 177. **Bronze or gun-metal**, an alloy of copper and tin, is the metal which is generally used for ordinary machinery; it is composed of 90 parts of the former and 10 parts of the latter, but for **very great pressures** the proportion is 86 of copper and 14 of tin, to 82 of copper and 18 of tin, the hardness of the alloy increasing with the quantity of tin.[5]

The hardness and resistance to wear is also increased by alloying some 2 per cent. of phosphorus with the tin and copper, producing a metal known as **phosphor bronze**.

[1] For explanation of how to set it out, refer to author's "Geometrical Drawing," p. 171.
[2] Refer to author's "Machine Design, Construction and Drawing," p. 265.
[3] Ordinary cast-iron bearings wear well if the velocity does not exceed 150′ per minute and the pressure 100 lbs. per sq. inch.
[4] Prof. Goodman.
[5] Conversely, to produce a soft, tough bronze, suitable for the teeth of wheels subjected to shocks, the proportion of tin is decreased, the usual parts being 8 of tin and 92 of copper.

EXERCISES.

Designing, etc.

1. Make a sketch design for a pair of brasses for a 3" journal, length 4½"; give dimensions. Figs. 427A and 427B type.
2. Design a frame bearing, fitted with brasses suitable for a 4" shaft, the length of whose journal is 6". Dimension the principal parts. Figs. 418 and 419.

Sketching Exercises.

3. Make simple sketches to show diagrammatically the forms of the following types of bearings:—bushed solid, single collar thrust, ordinary thrust block, and footstep.
4. Show by sketches six different ways in which the brasses of bearings can be arranged for *adjustment*.
5. Make a freehand sketch of Seller's self-adjusting pedestal.
6. Show by a sketch how brasses are fitted with white metal.
7. Sketch a wall bracket for a shaft pedestal.
8. Make a sketch of Schiele's pivot bearing. What advantage is claimed for this form, and what is its principal disadvantage? What is the name of the curve used?

Drawing Exercises.

9. Make working drawings of the adjustable hanger bearing, Figs. 427T and 427U, for a 3" shaft. Scale half size.
10. Make working drawings for the wall bracket, Figs. 428 to 430; diameter of shaft, 5". Scale 3" = 1 foot.
11. Draw two sectional elevations of the wall box, Figs. 435 and 436, suitable for a 4" shaft. Scale 3" = 1 foot.
12. Set out three views of the footstep bearing, Fig. 437; diameter of shaft, 3". Scale half full size.
13. Make a set of working drawings of the Plummer block, Figs. 416 and 417; diameter of shaft, 2½". Scale full size.
14. Draw three sectional views of Seller's self-adjusting pedestal, Figs. 422 and 423; diameter of shaft, 2". Scale full size.
15. Make working drawings of the 3" Plummer block (Fig. 427, Art. 173) to the instructions given.
16. Make the drawings of the crank shaft bearing (Figs. 424 to 426) in accordance with the instructions given in Art. 174.

CHAPTER XVIII

ROLLER AND BALL BEARINGS

185. Introductory Remarks.—Roller and ball bearings are used with the object of reducing the resistance due to friction, by substituting **rolling friction** for **sliding friction**, which is sound practice whenever it can be done without unduly sacrificing the simplicity, reliability, and comparative inexpensiveness of the ordinary type of bearing. But in dealing with such bearings we must not overlook the fact that with a plain bearing, perfectly made in its most efficient form, with a long, case-hardened, ground-steel journal, running in phosphor-bronze steps, and efficiently mechanically lubricated with an oil that possesses high lubricating properties without being very viscous, we have a journal practically floating on a film of oil, with practically only the small resistance due to **fluid friction**. On the other hand, in most cases it is not practicable to secure, or at least to maintain, such favourable conditions of running, with the result that the sliding solid friction which then occurs, heats and wears the parts which are in sliding contact,[1] and provision has to be made to take up this wear, which in bearings of this class is usually a simple matter. But it is a principle of the **roller** and also of the **ball** bearing that **wear should not take place**, *and no provision is made to adjust the bearings due to wear*. Of course, in bearings of this class that have been well designed for the pressure they have to support, the speed they have to run at, and made with great accuracy, no adjustment is necessary, or, rather, when signs of wear appear the worn parts are replaced with new ones. Such bearings, we shall see, under the best conditions of running, are very efficient, particularly when the machine to which they are fitted is being started from a condition of rest;[2] but, on the other hand, they are expensive to make and are liable to be seriously damaged by a crushed ball or fractured roller, as we shall see in dealing with the following representative types, commencing with the simplest application of rollers.

186. Rollers for Bridge Ends, etc.—To allow for free expansion and contraction, one end of a bridge usually rests upon nests of turned-steel or wrought-iron rollers, mounted to run between planed surfaces. Their length and number are usually decided by allowing a maximum load per lineal inch of each roller, of $600\sqrt{d}$ for steel, and $500\sqrt{d}$ for wrought iron, where d is the diameter of the rollers in inches, which should not be less than 2″.

[1] There can be no doubt that the loss of power in line and other shafting, with ordinary bearings, is often a very serious matter. For instance, it has been stated that at Sir Christopher Furness, Westgarth & Co.'s Engine Works, Middlesbrough, in one shop where two long lines of shafting and a comparatively small number of tools were used, *the loss was* 75·6 *per cent.*, and that in two other shops it was 59 and 42 per cent. Probably, under favourable conditions, it is rarely less than 20 per cent. It is fairly safe to assume that if the same amount of attention had been given to the perfection of roller bearings as has been given to ball bearings, we should see and hear a great deal more about them, as the saving in power and lubricants cannot be questioned, but, of course, the cost at present is against them. And we shall see that there are certain drawbacks peculiar to them.

[2] For this reason they have been fitted to electric railway carriages with advantage. It is claimed by the Empire Roller Bearings Co. that, with heavy rail vehicles, the frictional resistance in starting is reduced to rather less than 3 lbs. per ton, including the wheel friction upon straight, level rails.

187. Anti-friction Wheels.—Another simple roller arrangement is shown in Figs. 442 and 443. Heavy grindstones are sometimes mounted in this way, C being one end of the axle. Of course there is a certain amount of sliding friction in the bearings A and B, and it can easily be shown that—

$$\frac{\text{Friction with plain bearings}}{\text{friction with anti-friction wheels}} = \frac{d}{D \cos \theta}, \text{ or as } \frac{d}{D} \text{ nearly.}$$

To prevent end movement of the main axle, all the axles must be exactly parallel and the rollers the same size.

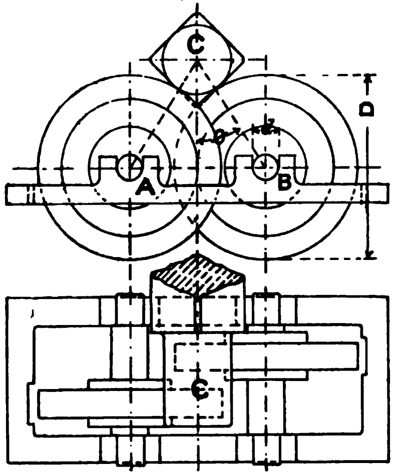
188. Roller Bearings.—In Figs. 444 and 445 we have shown the **simplest form** of roller bearing for a journal. Obviously, there is some friction between the rollers themselves in this case, and, as they cannot fill the annulus completely, it is possible for them to get slightly out of parallelism with the axis of the shaft, and then a **spinning**[1] or grinding action takes place, and *the line of contact with the journal is curved instead of straight*, which causes the roller to bend, or perhaps break, with disastrous results to the bearing.

189. Ring Cage Roller Bearing.—This type is shown in Figs. 446 and 447, and, although an improvement on the previous one, is apt to give trouble. It will be seen that the ends of the rollers are turned down to form small journals, which are carried in a pair of rings forming a **cage**, **frame**, or **yoke**; the shaft is fitted with a hardened and ground steel sleeve and the casing with a similar liner, both providing a hard and perfectly smooth path for the rollers, which are, in the best work, of tool steel, hardened and ground, or case-hardened. Although this arrangement of making all the engaging surfaces of hard steel gives the best service with the least wear, these bearings are sometimes made with the rollers running in contact with the cast-iron casing and wrought-iron or steel shaft, for light pressures and slow speeds.

190. Solid Cage Bearing.—Another arrangement is shown in Figs. 448 and 449, the rollers being carried in a solid gun-metal *cage* or *yoke* so that their axes always remain practically parallel with the axis of the shaft. Of course this means a certain amount of sliding between rollers and cage, but this does not appear to much reduce the efficiency. Bearings of this form have with advantage been used for the axles of electric railway carriages, and it is claimed for them that in starting or at such slow speeds that air resistance may be neglected, the driving effort is only about 10 per cent. that required by the same car fitted with ordinary bearings. The rollers are sometimes made in two or more lengths, breaking joints, as shown. In

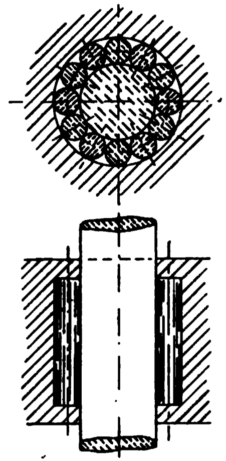
TYPES OF ROLLER BEARINGS.

ANTIFRICTION WHEELS FOR GRINDSTONE.



FIGS. 442, 443.

ROLLER BEARING SIMPLEST FORM.

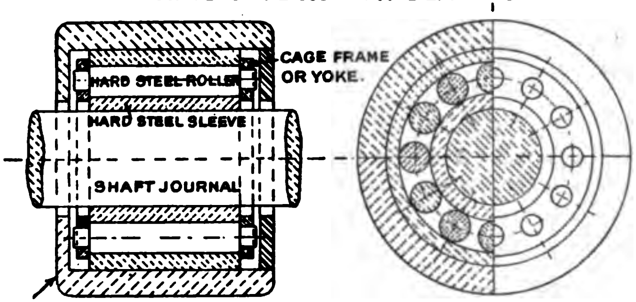


FIGS. 444, 445.

[1] Combined rolling and sliding, which, in the thrust bearing, Figs. 456 and 457, is analogous to what occurs between the pan and rollers of a mortar mill.

all cases such rollers should be made with convex ends and the edges rounded off to prevent them breaking away under pressure, as a broken roller is almost sure to cause rapid destruction of the bearing.[1]

Should any of the bearing surfaces lose their true cylindrical form, or be initially faulty in this respect, the roller will tend to twist out of line with the shaft and to have an end movement which causes **end thrust**, amounting, according to Professor Goodman,[2] to one-tenth of the load in some cases.



FIGS. 446, 447.—Bearing with hard steel casing or shell.

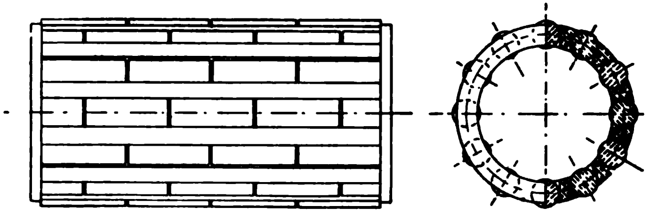


FIG. 448. FIG. 449.

191. Flexible Rollers for Bearings.—We have seen how important it is that the rollers should remain in contact with the shaft along their entire length. Now, this condition is much more likely to be satisfied if the roller is made *flexible*, even if the surfaces upon which it rolls are not exactly true. Fig. 450 shows such a roller manufactured by the Hyatt Roller Bearing Co. It is made by winding a steel bar or ribbon of rectangular section about a mandril.[3]

Another ingenious flexible roller, due to Messrs. Kynock, is shown in Fig. 451. It is made by rolling a suitably shaped steel plate into cylindrical form, giving a helical division along it as shown.

192. Conical Roller Thrust Bearings.—We have seen (Art. 182) that the end thrust of a shaft or other part is resisted by either a step bearing or collar bearing. Now, by the use of suitable rollers, arranged in accordance with geometrical principles, it is easy to construct thrust bearings in which the friction is due to rolling action instead of sliding, giving a considerable increase of efficiency. Fig. 452 will assist in making clear the conditions that must be satisfied. The rollers A and B are truncated cones, whose common vertex is in the axis CD of the shaft at C, and when so constructed there will be a true rolling action between the parts without slipping. In this case the bed EF is flat, and the end of the shaft conical, but, obviously, these could be reversed; indeed, it is sometimes convenient to do this. Fig. 453 shows a case where the axes of the rollers are at right angles to the axis of the shaft, the common vertex of the rollers being at C in the axis DC; in fact, so long as this condition

[1] Properly designed bearings permit of very great pressures being carried. Indeed, it is claimed by the Mossberg Roller Bearing Co. that in rolling mill practice these bearings will permit of a pressure of 20,000 lbs. per square inch on the projected area of the journal. Messrs. Mossberg also cite a remarkable example of the efficiency of their roller bearings. A *steel* wheel weighing 130 lbs., and 14" diameter, was speeded up to 10,000 revolutions per minute, and continued revolving for one hour and thirty-three minutes after suddenly disconnecting the source of power, the test being repeated forty times with no detriment to the bearing. An interesting and appropriate application of roller bearings was made when the Empire Roller Bearing Co. mounted the huge bell (weighing 23 tons), **Great Paul**, on their bearings. The friction was found to be only one-seventh what it was with the former bearings.

[2] Goodman's "Applied Mechanics," p. 243, in which Professor Goodman says that he has not found any roller bearing entirely free from "end thrust." Sometimes the friction of the bearing diminishes as the speed increases.

[3] These bearings are used on some motor-cars, particularly those of American make, the Ford car being fitted with them.

is satisfied, the angle may be varied to best suit any given case. To guard against any tendency of the rollers to be forced out radially, a *retaining cage* is required. This is shown in position in Fig. 454, the plan and sectional elevation of the cage, showing the rollers in position in the *pockets* of the cage arranged to receive them with a good working fit, the pockets restraining them from twisting round out of position so that their axes would not intersect that of the shaft.

In heavy work a ball is fitted to the large end of each roller, as shown in Fig. 455, to reduce the friction due to any **end thrust**.

It can be shown [1] that the angle at the vortex of the roller's cone should in no case exceed 15°. They are frequently made from about 10° to 14°, as shown in Fig. 452. When accurately made and running on steel,[2] the **frictional resistance** is **very small**, especially **at low speeds**, the bearings often having a coefficient of friction less [3] than 0·003.

HYATT FLEXIBLE ROLLER.

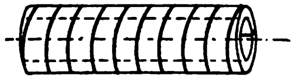


FIG. 450.

KYNOCK FLEXIBLE ROLLER.

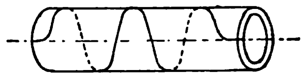


FIG. 451.

193. Cylindrical Roller Thrust Bearings.—For **very heavy work** these bearings are more often used than those with conical rollers.

ROLLER THRUST BEARINGS.

BEARING SURFACE FLAT

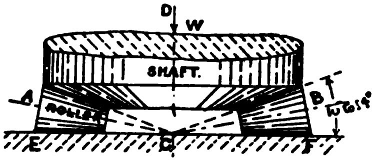


FIG. 452.

BEARING SURFACE CONICAL.

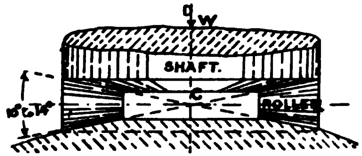


FIG. 453.

THRUST BEARING COMPLETE.

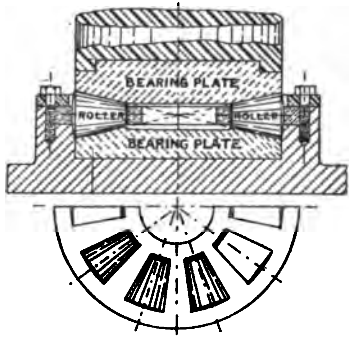


FIG. 454.

THRUST BALL IN ROLLER

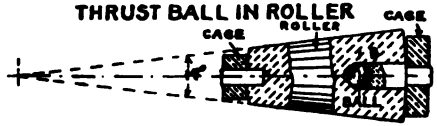
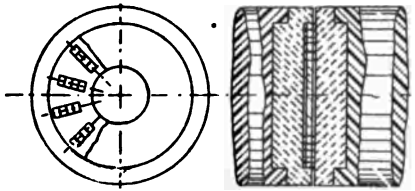


FIG. 455.



FIGS. 456, 457.—Cylindrical roller thrust bearing.

The bearing plates are flat, as shown in Figs. 456 and 457, and several narrow rollers all the same size are put in each pocket side by side, the sets being arranged at different distances from the shaft's axis to minimize the tendency to groove the bearing plate. The

[1] Refer to the author's "Machine Design, etc.," p. 109.

[2] Wrought-iron rollers on cast iron may perhaps be used for the lightest pressures at slow speeds, but for satisfactory running, particularly with heavy pressures at high speeds, the rollers and the bearing surfaces upon which they run should be tool steel, hardened and ground to form. Mild steel, case-hardened, has also been successfully used.

[3] A case is recorded in *Cassier's Magazine*, May, 1897, where, with a working pressure of 104·5 lbs. per square inch, μ the coefficient of friction was = 0·0025.

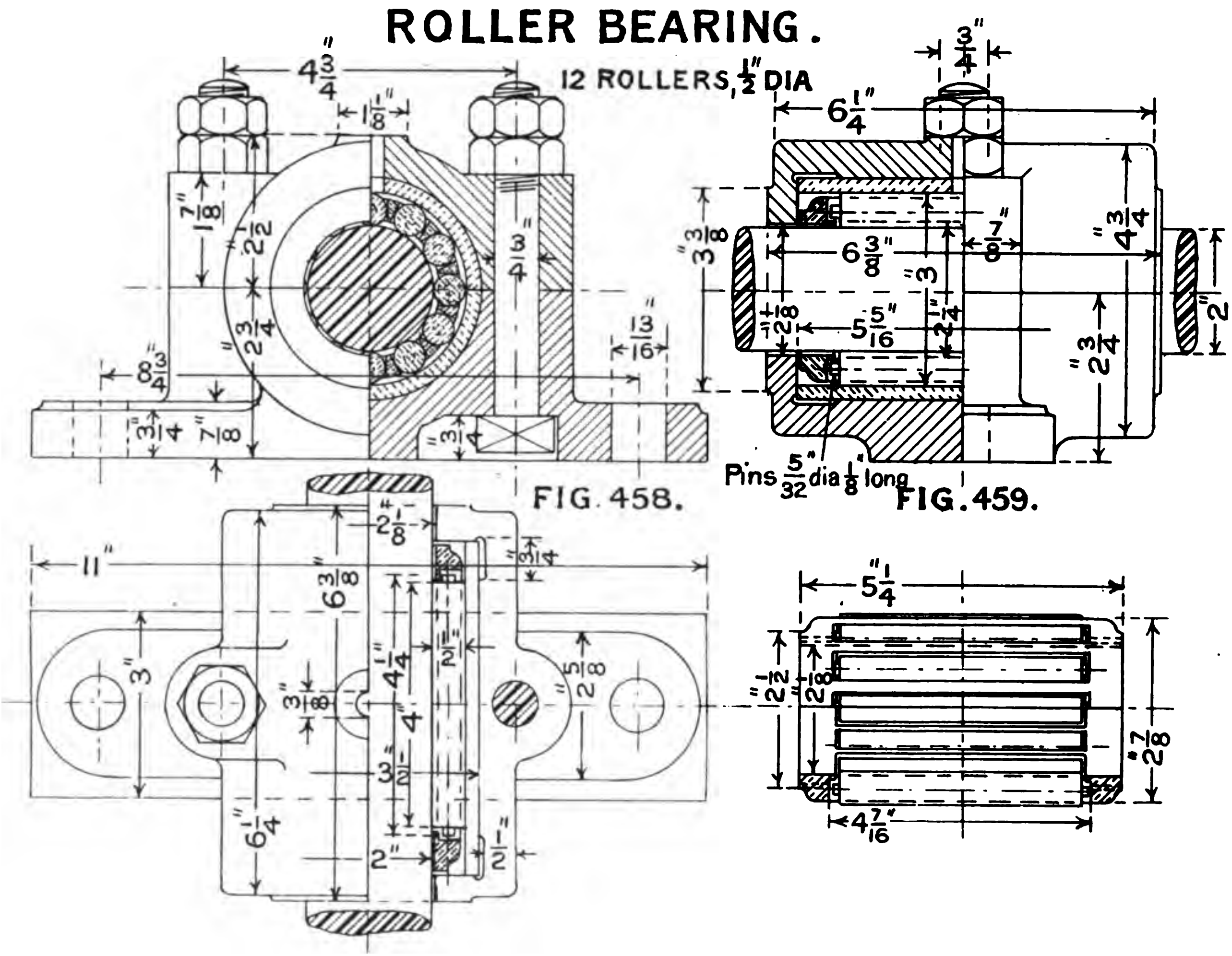


FIG. 458.

FIG. 459.

FIG. 460.

FIG. 461.

ends of the rollers are slightly crowned and the axes are radial to the bearing. As each roller is carried round the axis of the shaft it travels a distance equal to the circumference of a circle whose radius is the mean distance of its ends from the axis, and it also **spins** around once in relation to the plate, but, with the very short rollers (often only ½″ long and ½″ diameter) generally used, this *grinding* of the rolling surfaces apparently has little effect.

194. Drawing Exercise.—In Figs. 458 to 461, we have a fully dimensioned 2″ roller bearing, which can be conveniently used as a drawing exercise: the four views may with advantage be drawn full size. They should present no difficulty to a first year's student.

For *line shafting* these pedestals should be made with *swivel seatings.*

195. Ball Bearings.—In Art. 185, we discussed several matters that applied equally to roller and ball bearings, but, in dealing directly with the latter, we may first endeavour to get clear ideas as to the conditions which must be satisfied if we are to secure pure rolling motion (or something closely approximating to it) in various forms of ball bearings. Fig. 462 shows a longitudinal section of the simplest form of ball bearing freed from all auxiliary parts; it is an example of pure rolling motion on cylindrical

EXAMPLES OF BALL BEARING CONTACTS.

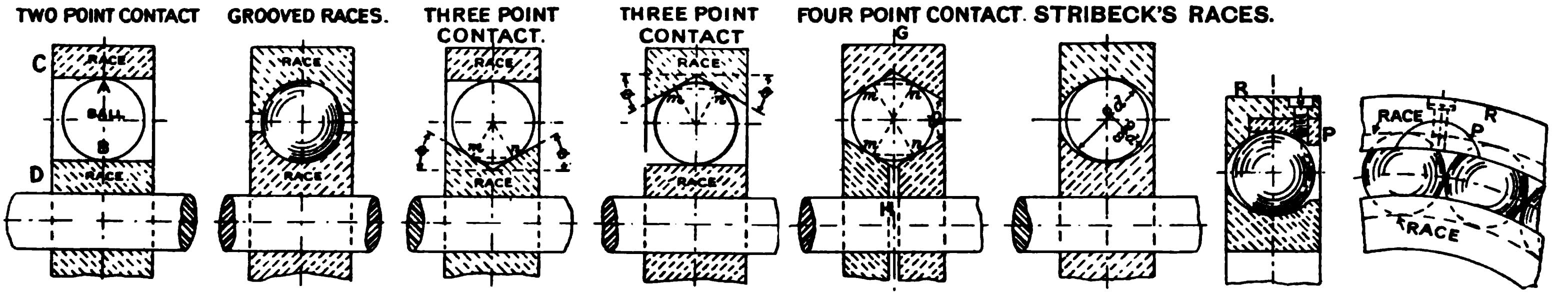


FIG. 462. FIG. 463. FIG. 464. FIG. 465. FIG. 466. FIG. 467. FIG. 468. FIG. 469.

surfaces, and it is called a **two-point bearing,** because contact occurs at two points, A and B, and the parts C and D, which are in rolling contact with the balls, are called **races,** each race being a solid ring. Now, to prevent any tendency of the balls to get out of position sideways, it might appear that the best form of the races would be grooves whose radius is just equal to that of the balls, as shown in Fig. 463, but, obviously, with such an arrangement the friction becomes excessive; so, to avoid this, the three- or four-point contact is used. Figs. 464 and 465 are two examples of **three-point contact**; in each case one of the races is cylindrical and the other grooved, both sides of the groove making the same angle with the axis of the bearing, so that *mn*, Fig. 464, the two points of contact of the ball against the sides of the race, shall be the same distance from the axis (this also applies to Figs. 465 and 466). It can be easily understood that when the ball is pressed between the sides of the groove the pressure will slightly flatten the ball where the contacts occur, instead of the contacts being geometrical points, as they would be if the ball and race were *inelastic.* Now, this being so, it can be conceived that the ball has a rolling motion in the groove combined with a slight **spinning**[1] motion

[1] We have referred to this spinning or grinding action in Art. 188.

about the axis *mn*. Of course, with this form of bearing, the only resistance it can offer to **end motion** (in the direction of the shaft's axis) is the sliding friction between the balls and the cylindrical surface; on the other hand, care must be taken to make the angle θ large enough or the balls may bind or wedge between the races and probably become fractured.

In Fig. 466 is shown a **four-point contact** arrangement. Although this bearing is chiefly used to support a pressure normal to the axis of rotation, it will also resist *to a certain extent* end thrust, but *two firmly fixed bearings of this kind should not be used on the same shaft*, especially when it is a long one, as any change of length of the shaft due to differences of temperature tends to force the balls against the sides of races with possible damage to them or the balls. Such a bearing, however, can be used on the same shaft with others made a loose fit in their respective **housings** so that they do not restrain end motion. The angle θ should be not less than 30°, and this also applies to θ in Fig. 466. Fig. 467 shows the form of races (first suggested by Professor Stribeck), which are now largely used, they are struck with a radius of about $\frac{9}{8}$ to $\frac{11}{8}$ times that of the balls. With this form a greater load can be carried with less friction. Figs. 468 and 469 show how one of the rings or races R may be fitted with a removable piece P, accurately fitted, and held in position by the screw shown, to allow of the balls being introduced or withdrawn.

196. Journal Hub Ball Bearings. Form of Constraining Surfaces.—In arranging the constraining surfaces care must be taken that the balls have no effective tendency to leave their proper path; thus in Figs. 470 and 471, the tendency of the balls to leave the races is reduced to a minimum, as the races are so formed that true rolling occurs. When a load W (Fig. 470) on the **three-point contact** bearing is supported, the reactions Q and R (whose relative magnitudes are shown in the triangle of forces) at *a* and *d* on the hub and cone respectively, create small areas (as we have seen) on the ball and cause it to roll with a motion akin to that of a cone, whose sides *d*C and *ba*C *should intersect in a point* (C) *on the axis* CF, as shown, if **true rolling** is to occur. To avoid any tendency of the balls to wedge between the cup and cone, the angle at A should not be less than 30°. Fig. 471 shows the arrangement for a **four-point contact** bearing. It has the advantage of being more compact than the preceding one. Of course, the angle at A, and the corresponding one opposite, should not be less than 30°, and the lines *ab*C and *ed*C (passing through the contact points) must intersect in the axis, as shown at C, and just explained. The well-known **cup-and-ball two-point contact** is shown in Fig. 472: it is extensively used for the wheels of cycles and very light cars, etc.; each ball runs in a pair of concave races, whose radius should not exceed some $\frac{11}{8}$ the radius of the balls, to prevent side motion, for with this arrangement **the balls are not in a stable condition**, the actual positions of their contacts with the races being indeterminate. Of course, any tendency of the balls to roll further out from the axis or nearer to it is resisted by the increasing slope of the sides of the races, so the bearing automatically adjusts itself into a position of equilibrium, but to prevent wedging it is advisable to well lubricate the bearing. Obviously, this bearing is **capable of resisting a small side or end thrust**, but we shall directly see that for heavy vehicles, where very considerable side thrusts occur, a more satisfactory bearing is used.

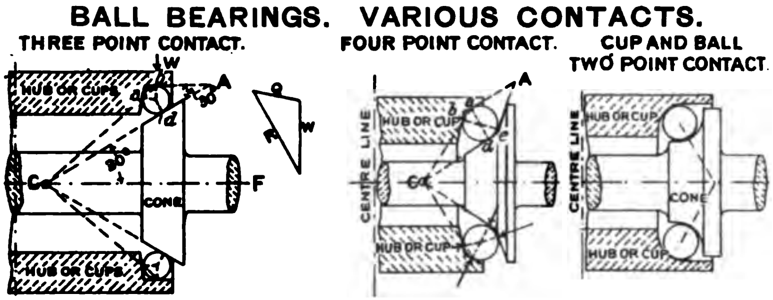


FIG. 470. FIG. 471. FIG. 472.

197. Ball Thrust Bearings.—The simplest way of taking the end thrust or pressure of a shaft is to arrange balls between two rings or discs whose planes are normal to the axis of motion, but this necessitates the use of a retaining cage, as we shall directly see. If this is to be avoided, one or both of the rings or discs may be grooved to constrain the balls to move in a circular path. Fig. 473 shows one way of doing this, the lower or bearing ring being flat and the upper ring grooved, giving a **three-point contact.** As explained in the previous article, if **spinning** or **grinding** of the balls is to be prevented, the sides of the grooves must be so shaped that a line *baC*, through the points of contact *a* and *b*, must cut the surface C*m* of the ring in a point C in the axis of motion, as the motion of the balls is akin to that of a cone, as previously explained. The pressures on the balls due to a weight W is shown by the triangle of forces, WQR. One of the rings may be coned, as in Fig. 474, at DE. Then, if DE be produced till it cuts the

BALL THRUST BEARINGS.

THREE POINT CONTACT THRUST.

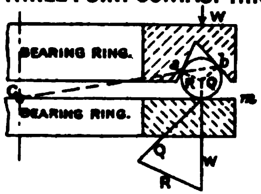


FIG. 473.

THREE POINT CONTACT THRUST.

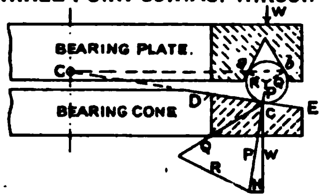


FIG. 474.

FOUR POINT CONTACT THRUST.

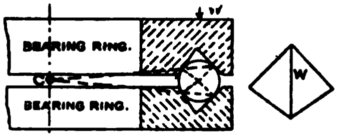


FIG. 475.

BALL THRUST WASHER
TWO POINT CONTACT WITH CAGE.

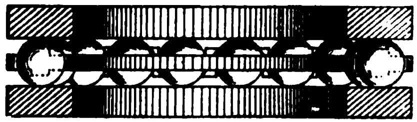


FIG. 476.

axis in C, a line C*ab* must cut the ball in *a* and *b*, the contact points with the sides of the groove, giving a three-point contact. In this case W is resolved into the normal force P acting on the ball and a force H perpendicular to the axis, P being resolved into R and Q, the other two pressures on the ball. If both rings be grooved, we get **four points of contact,** as in Fig. 475, which should speak for itself. Professor Goodman has called attention to the fact that "although these forms of thrust bearings are right in principle, they are not found to work well in practice, probably because the exact conditions are upset when any wear or change of load takes place. A series of tests of some bearings of this type showed that the balls began to *peel* and score and the races to grind at very low loads and speeds." Hence, we find in the best practice that flat or slightly hollow races are used, giving a two-point contact, as shown in Fig. 476, which represents one of Messrs. Hoffmann's **ball thrust washers** [1] designed for the spindles of drilling machines, feed and elevating screws, worms and mandrils of lathes, etc. The balls are held in the gun-metal retaining case or ring [2] which keeps them in position and prevents them falling out when the bearing is dismantled. The diameters of the holes in the top and bottom hardened steel washers which form the *ball races* are made five thousandths of an inch above standard size, so as to allow the shaft to revolve freely inside the *stationary* race.

198. Ball Journal Bearings.—The simplest form of a single-row ball bearing is shown in Fig. 477. It can only be used when a

[1] These bearings carry no *journal* pressure, but only end thrust. To equally distribute the load over all the balls, one of the washers is sometimes fitted with a spherical back which allows it to swivel into its exact position when the load is applied.

[2] The thickness of the ring is about one-half to two-thirds the diameter of the balls, and holes are drilled for the reception of the balls, the top and bottom edges of the holes being slightly burred over to keep the balls in position.

journal load has to be carried; great care must be taken to fix the shaft and bearings, so that the former may be free to expand and contract, due to **differences of temperature,** without putting any end thrust whatever upon the bearing. The most satisfactory way of fixing the internal race ring is to make it conical, as in Fig. 478, and to hold it on to a corresponding conical part of the shaft (or sleeve) by means of a nut or collar screwed upon the shaft, or, if this cannot be done, a split conical sleeve should be drawn into the conical part of the race ring by a collar nut or back nut to grip the shaft and race, a set-screw being used to prevent the collar-nut working off, as shown in the figure (478).

HOFFMANN'S BALL BEARINGS.

Fig. 477.—Single row journal bearing.

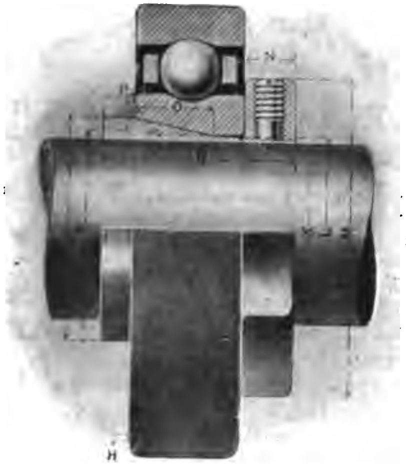


Fig. 478.—Journal bearing.

Fig. 479.—Single compound.

199. Compound Ball Bearings.—When a bearing is constructed by combining the **thrust and journal bearings** we have referred to, it is called a *Compound Bearing.* If arranged as in Fig. 479, it is a *Single Compound Bearing,* but of course this can take the **thrust in one direction only,**[1] and is therefore suitable for such arrangements as footstep bearings of vertical shafts, for clutch shafts of motor-cars, where conical clutches are used and there is an end thrust, etc. Referring to the figure just above and to the right of the journal balls, a spring will be seen; it is slightly in compression, and keeps the balls properly in contact with their races, it also allows of any slight contraction or expansion of the shaft. In certain applications of this bearing the spring is dispensed with, and a distance piece used to fill space occupied by it. It is absolutely necessary to firmly clamp the cone upon the shaft, and the simplest way of doing this is shown, it being a slight variation of the one explained in the previous article. When it is required to remove the bearing from the standard housing, the small locking screw is removed and the disc nut unscrewed at the right-hand end of the housing, where the bearing can be withdrawn. The balls, being held in ball retaining cages, will not drop out.

Students may refer to the author's "Machine Design, etc.," p. 283, for data for designing ball bearings.

[1] It is often convenient to use two of these *single* compound bearings on the same shaft, in cases where there is thrust in both directions, instead of a *double* compound one.

EXERCISES.

Drawing Exercise.

1. **Make working drawings (four views) of the roller bearing shown in Figs. 458 to 461. Scale full size.**

Sketching Exercises.

2. Make sketches showing the difference between a *ring cage* and a *solid cage* for a roller bearing.
3. Show by sketches two ways of arranging a roller *thrust bearing*. In one case the bottom bearing plate is to be flat, and in the other conical.
4. Make a sketch of a cylindrical roller thrust bearing. Why are the rollers in such bearings usually staggered?
5. Explain what is meant by a roller or ball *spinning*. Illustrate your answer by sketches. Do you consider ball bearings require lubricating? If so, why, and what lubricant would you use?
6. Show by diagrammatic sketches the difference between three-point and four-point contact in *ball bearings*. What conditions must be satisfied if spinning is not to occur?
7. Show by a diagrammatic sketch the difference between three-point and four-point contact in ball *thrust* bearings, and define the conditions which must be satisfied if there is to be true rolling contact on each of the races.

CHAPTER XIX

TOOTHED GEARING

200. Introductory Remarks.—Toothed wheels have been used for the transmission of motion and power since the days of Archimedes, about two centuries before the Christian era, but it remained for geometricians of comparatively recent times to investigate and solve the problems which have enabled the engineer to shape the teeth of wheels so that practically the same *uniformity of motion*[1] can be transmitted from one to the other as if they were plain cylinders (or cones) *rolling* on one another by frictional contact; in fact, in every pair of spur wheels we have two imaginary cylinders[2] (and in every pair of *bevel* wheels two imaginary cones) provided with certain projections or *teeth*, and intermediate depressions or tooth *spaces*, so that the teeth of one wheel enter the spaces of the other, but, when these teeth are properly formed, the velocity of one wheel in relation to that of the other, that is, the *angular velocity ratio*, is inversely proportional to the diameters of the imaginary cylinders[3] and cones, as we shall see directly, and these cylinders and cones are represented by circles (called *pitch circles*) on the wheels, as shown in Figs. 480 and 481.

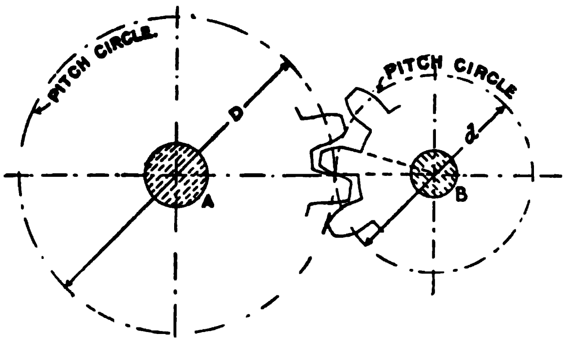


FIG. 480.

201. Relative Speeds.—If the pitch circles, Fig. 480, roll on one another they have the same velocity, that is, travel the same distance in a given time; therefore $ND\pi = nd\pi$, or $\frac{D}{d} = \frac{n}{N}$, where N and n are there volutions per minute of wheels A and B respectively. But the angular velocity is proportional to the revolutions; therefore *the angular velocities are inversely proportional to the diameters of the pitch circles.*

202. Technical Names of Teeth Details.—An inspection of Fig. 481 will enable the student to understand the meaning of the technical names of the different parts of wheel teeth, as we shall frequently have to make use of them. The diagram should speak for itself.

203. Pitch, etc.—Fig. 481 shows that the *pitch* of a wheel is the distance, centre to centre, of two adjacent teeth, but there are

[1] As the inertia of heavy moving parts resists alteration of velocity, any variation in the uniformity of the motion causes the driven wheel to alternately fall back and overtake the driving wheel (a jerky action, technically called back lash), with loss of power, vibration, and noise. Wheels correctly designed, made, and fitted, will bear evidence of uniform contact from the point of *each tooth* to some distance below the pitch circle.

[2] The pitch surfaces.

[3] This condition of constancy of velocity ratio may to some extent be secured by having the teeth small and numerous.

two ways of measuring it, namely, along the **arc,** and along the **chord,** and among the earlier authorities there used to be a great difference of opinion as to which of these is correct. But, if the diameters of wheels are to be exactly proportional to the number of teeth, the pitch must be measured by the length of the *arc*, or along the curved pitch line, and this, the obviously correct way, was adopted by Willis, Rankine, and others, and is now the accepted mode. This being so, the relationship of *circular* pitch,[1] diameter, and number of teeth is expressed by the equation $D\pi = Np$, where D = the diameter of pitch circle in inches, p = pitch in inches, and N = the number of teeth.[2]

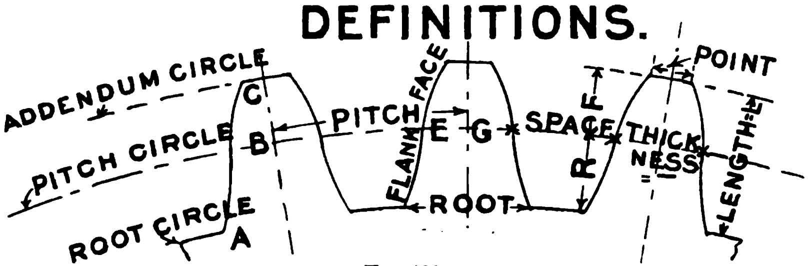


FIG. 481.

Then $$D = \frac{Np}{\pi},\ p = \frac{D\pi}{N},\ \text{and } N = \frac{D\pi}{p}$$

204. Diametral, or Manchester Pitch.—The use of this pitch has been much extended since the advent of the motor-car in this country, as it has been largely used in connection with its gear wheels (the other pitch used for these wheels is the French **Module**), thereby avoiding inconvenient fractions in their pitch diameters, as with the *diametral pitch system* the diameters of the pitch circles can always be made suitable. In the practice of Messrs. Browne and Sharpe, and Messrs. Sharp, Stewart, & Co., of America, **the dimetral pitch equals the number of teeth divided by the diameters of the pitch circle.**[3] So that **it is a ratio and not a measure** like the circular pitch. To further explain, let D = the diameter of pitch circle, p_d = the diametral pitch, N = number of teeth, p = circular pitch.

Then $$p_d = \frac{N}{D},\qquad N = Dp_d,\qquad D = \frac{N}{p_d}$$

And **diametral pitch number** $= \dfrac{1}{p_d}$

The addition to diameter for increased number of teeth $= \dfrac{\text{Number to be added}}{p_d}$

Outside diameter of wheel $= \dfrac{2}{p_d} + D.$

And circular or true pitch $= \dfrac{\pi}{p_d}$

[1] *Circular pitch* must not be confused with *diametral* pitch. When the word "pitch" is alone used, circular pitch is referred to.

[2] It is the practice of some engineers to make one of a pair of equal wheels with an additional tooth called a **hunting cog**. Then each tooth of one wheel will encounter each tooth of the other equally often, and the wear will be equalized. Any pair of wheels will have a *hunting cog* if the teeth of both cannot be divided without remainder by any number except 1. In other words, the numbers must be *prime* to each other.

[3] But the **diametral pitch** which is perhaps more commonly **used in this country** is the reciprocal of this, or $\frac{D}{N}$. Then the circular or true pitch $p = \pi p_d$.

Distance between axes, or centre distance $= \dfrac{\text{Sum of the number of teeth}}{2p_d}$

EXAMPLE.—A 12 pitch wheel (diametral, or Manchester pitch), 8″ diameter, will have $12 \times 8 = 96$ teeth, and the true or *circular pitch*

$$= \frac{3\cdot1416}{12} = 0\cdot2618''.$$

205. Module, or French Pitch.—This pitch, which we have just referred to, is the one in general use in France, the country from which so many of our finest motor-cars have come, so that the business relations between the two countries in connection with this remarkable industry, to say nothing of the inherent advantage of the system, are leading to an increasing use of it in this country. **The module is the diameter of the pitch circle in millimetres divided by the number of teeth**, and it equals the length of the face part (M) of the tooth, as shown in Fig. 481A. For other relationships—

MODULE OR FRENCH PITCH.

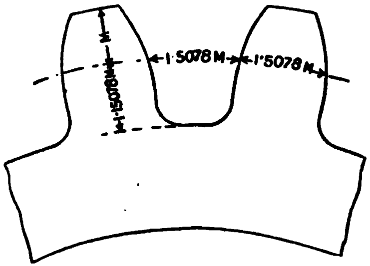


FIG. 481A.

Let D = diameter of pitch circle in millimetres.
D_2 = diameter of circle tipping teeth.
M = **module** in millimetres (face length or height above pitch line).
N = number of teeth.
p = circular pitch in millimetres.
T = thickness of teeth on pitch line.
x = clearance at top of teeth.
Then root length = M + x.

And

$$M = \frac{D}{N} = \frac{p}{\pi} \qquad \therefore p = \frac{\pi D}{N} = M\pi. \qquad \text{And } x = \frac{1\cdot5708M}{10}$$

$$D = MN \qquad \therefore N = \frac{D}{M}. \qquad D_2 = M(N+2) \qquad \therefore N = \frac{D}{M} - 2.$$

206. EXAMPLE.—A wheel with 60 teeth and Mod. 10 pitch will have a *diameter of pitch circle* = $10 \times 60 = 600$ mm., and an *outside diameter* of $10(60 + 2) = 620$ mm., whilst M, the face or height of the teeth above pitch line $= \frac{600}{60} = 10$ mm.

207. Form of the Teeth.—Geometricians have shown that there are only two curves, namely the **cycloid** and the **involute**,[1] which completely fulfil the condition of giving the perfect uniformity of motion referred to in Art. 200. Teeth of the cycloidal type are generally used in ordinary work, as they cause less thrust on the bearings than involute ones, the thrust being perpendicular to the line of centres at the instant of crossing it. On the other hand, the thrust of the latter is constantly in the direction of the common tangent of their bases. However, they have many advantages over the former, which make them suitable for use in some cases, particularly where the distance apart of the centres requires to be variable, as in rolling mills, or where **great strength** and practically **no back lash** are important factors, as in motor-car gears.

[1] Refer to the author's "Machine Design, etc.," p. 297.

We will now deal with cycloidal curves so far as they apply to the simple problems of forming the teeth of wheels. Now, if a circle be constrained to roll on another, any point in the moving circle (called the **rolling circle**) will describe a curve. If the rolling circle roll outside the other, as shown in Fig. 481B, which should speak for itself,[1] or as at P on circle B, Fig. 482, the curve PP_2 generated or described will be an **epicycloid**[2] (used for the faces of the teeth on wheel B), whilst if it roll inside circle A, as at P_3, the curve P_3P_4 is called an **hypocycloid** (used for the flanks of the teeth on wheel A). It is a peculiar, and in this connection useful, fact that, if the rolling circle be half the diameter of the circle it rolls on, the hypocycloid is a straight line, in fact, a diameter of the pitch circle. This is shown both at P_3P_4 and Q_3Q_4 in Fig. 482. It is often convenient to use this particular form of the curve for the flanks of the teeth, as they then become radial, as at B, Fig. 487. If, on the other hand, the *rolling circle* **rolls on a straight line**, the curve generated is a **cycloid**, used for the curves of the teeth of **racks**.

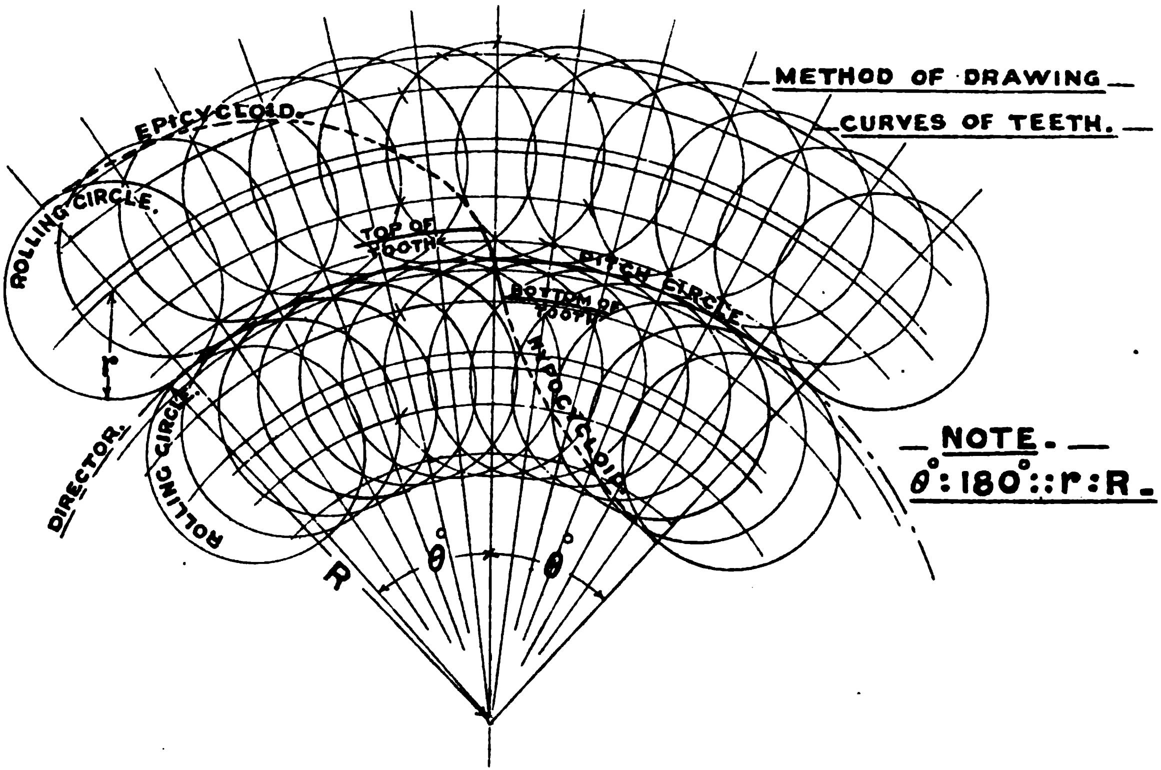


FIG. 481B.—Generation of cycloidal curves by rolling circles.

Now, let us suppose that A and B, Fig. 482, are the pitch circles of two wheels which are to gear together, we may, for present purposes, arbitrarily select

[1] Refer to author's "Elements of Geometrical Drawing," p. 182.
[2] For much useful information relating to these curves, see the author's "Elements of Geometrical Drawing," p. 177.

any size rolling circle to generate *all the curves*, but it will be better understood directly that, for practical reasons, this rolling circle must not in any case be smaller in diameter than the radius of the pitch circle of the smallest wheel in the train, in this case B; and if condition be satisfied for any number of wheels in a train, it is a fundamental fact that any two will correctly gear together, if the teeth be made of the *same pitch*. It will be observed that in all cases the part of the tooth below pitch line works only with the part above pitch line in its fellow, and *vice versâ*. This being so, it is obvious that we may elect to use a certain size rolling circle for the flanks of one wheel of a pair, and the same circle for the faces of the other, and this will enable us to have **radial flanks in each wheel**, which has been done in Fig. 482, the hypocycloid P_3P_4, and the epicycloid PP_2, being generated by circles of the same diameter, equal to the radius of pitch circle A. Also the hypocycloid Q_3Q_4, and epicycloid VV_2, are generated by circles whose diameter is the radius of the pitch circle B.

USE OF TEMPLETS IN SETTING OUT CYCLOIDAL TEETH.

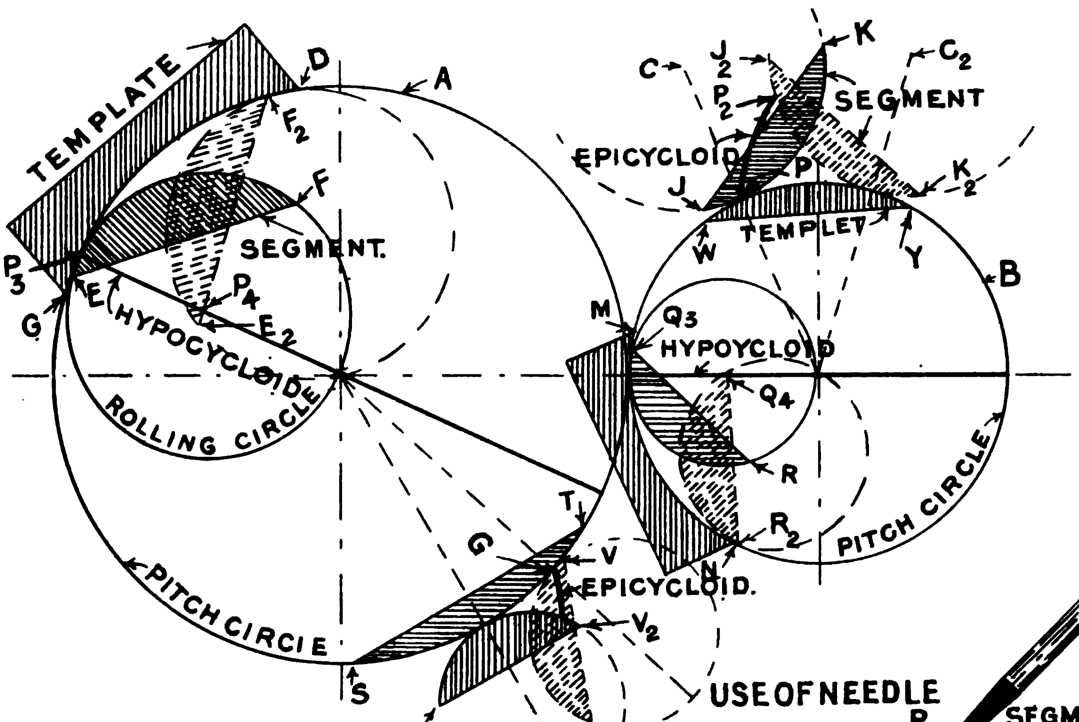


FIG. 482.

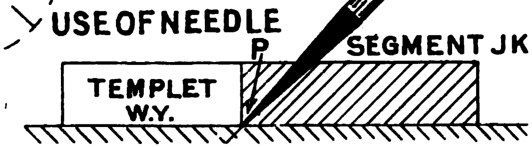


FIG. 483.

208. Setting out Cycloidal Teeth, Use of Templets, etc.—In setting out these curves or drawings of tooth forms the draughtsman

who has studied practical geometry experiences no trouble, for after geometrically finding a few points[1] in each curve, he draws a fair line through them, and then finds the centres and radii of circular arcs that closely approximate to the cycloidal curves, and uses them to describe the teeth; or **Willis' Odontograph**[2] can be used with advantage to find the centres of arcs which will closely approximate to the true cycloidal ones, so that all wheels of the same pitch will truly gear with one another. When this instrument is not used a portion of the pitch circle may be drawn to full size on paper (or on a smooth chalked board), and a **templet**, WY, Fig. 482, may be made, formed of wood, say $\frac{1}{8}''$ thick, shaped to the arc, and laid upon it. In the same way a **segment** of the rolling circle, JK, may be made, and a needle driven obliquely through the edge of it, as shown in Fig. 483, its point being made to just coincide with the arc at P. Then, by placing one hand on the *templet*, and with the other moving the *segment* (being very careful to prevent slipping), a clean fine line or groove[3] is made on the paper from P to P_2. In this case the same size rolling circle generates P_3P_4, as previously explained, but as this is a straight line, and diameter, it is drawn with a straight edge, and Q_3Q_4 is drawn in the same way, although for both of these the templets are shown in position, to illustrate the principle.[4]

We have seen that the **form of the root of the tooth** is influenced by the size of the rolling circle in relation to the pitch circle,[5] but Figs. 484 to 486 and Fig. 487 should make this matter still clearer. It will be seen that the root of tooth B, Fig. 487, is due to the straight line hypocycloid, Fig. 485, whilst with the smaller rolling circle of Fig. 484, we get the stronger root A, in Fig. 487, and the weaker root C, Fig. 487, with the larger rolling circle of Fig. 486.

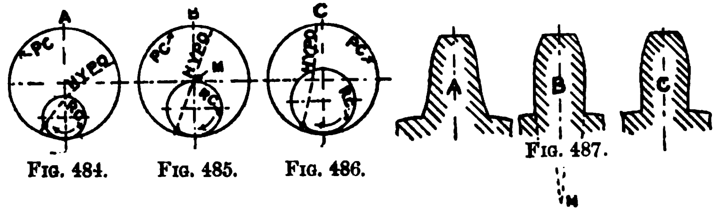


FIG. 484. FIG. 485. FIG. 486. FIG. 487.

209. Rack and Pinion. *Case (a)* **Rack working with Single Pinion.**—In this case the *faces of the rack teeth are* **cycloids**, generated by a circle half the size of the pitch circle of the pinion. And the *flanks of the rack* are **hypocycloids** whose generating circle is of infinite radius, and therefore they are *straight lines perpendicular to the pitch line.* So the **faces** of the teeth of the **pinion** must be **epicycloids**,

[1] Refer to the author's "Elements of Geometrical Drawing," p. 183. The circle BD there shown can be replaced by one cut out of stiff paper, with equal divisions marked on its edge, equal to those on the arc (pitch circle) EBK. Then, by rolling the paper circle and using the divisions for position points, points in the curve can be pricked off.

[2] This system is founded on true geometrical principles, first suggested by Euler, and worked out and perfected by Willis. For description of the instrument manufactured by Messrs. Holtzafell, see *Transactions Inst. C.E.*, vol. ii., 1838. Also, Willis's "Mechanism," 2nd edition, p. 120, or Fairbairn's "Millwork," vol. ii. p. 30.

[3] Care must be taken to fix the needle at such an angle that it will not *scratch* the paper; a fine indented mark is required.

[4] Having described the curves, arcs of circles are found with the compasses to nearly coincide with them. Then, if it is a matter of setting out the curves on the fronts of teeth of a pattern, or on the rough cogs of a wheel, a circle which is concentric with the pitch circle and passes through the centres of the arcs is described, and the compasses are adjusted to the radius of the arc, and by always keeping one point in the circle just described, two arcs are drawn for the face of each tooth, one to the right, the other to the left, and the operation is repeated for the flanks, having previously divided the pitch circle into equal parts corresponding to the pitch, and the thickness of the teeth having been marked on the cogs. Then these arcs serve as guides in shaping or finishing the acting faces.

[5] The teeth of all wheels of the same pitch that are to gear together, must be generated by the same *rolling circle*, the best diameter of which is $2{\cdot}22p$, unless any wheel in the set has less than fourteen teeth, in which case the diameter of the rolling circle should be $d = \frac{Np}{2\pi} = 0{\cdot}159Np$, where N is the number of teeth, in the smallest wheel. The *smallest number of teeth* that can be made to give more or less satisfactory proportions to the teeth of cycloidal pinions is eleven. The diameter of the rolling circle is then $d = \frac{11 \times p}{2\pi} = 1{\cdot}751p$. But the minimum number should be fourteen whenever practicable.

generated with a circle of infinite radius; and consequently **involute.**[1] The **flanks** of the pinion's teeth are **hypocycloids**, whose generating circle is half the pinion pitch circle.

Case (*b*) **Rack working with Set of Wheels.**—In this case the **flanks** and **faces** of rack teeth are **cycloids**, and the **faces** and **flanks** of the **pinion** teeth **epicycloids** and **hypocycloids**, respectively, the diameter of the rolling or generating circle for all the curves being $= \frac{Np}{2\pi}$.

209a. Arc of Action. Arcs of Approach and Recess.—Let us suppose that the rolling circle, centre C_2 (Fig. 488), rolls on the pitch circle B, and that E is a *generating point*, it will describe the hypocycloid FEH; but if it had rolled on pitch circle B_2 the epicycloid EG would have been described. Now, it should be clear that if the two pitch circles roll on one another E will coincide with G (at the pitch P) when they reach the line of centres CC_4; conversely, these points in moving back from P, in the direction of the arrow, will represent what occurs when the driving tooth EG drives the driven tooth EF from contact at P to contact at E (where the rolling circle cuts the addendum[2] circle), at which point *contact ceases*, and the arc PE is called the **path of recess**, *the point of contact of the two curves being always on the circumference of the rolling circle.* Similarly, we have at A_2 a second pair of teeth in contact at that point (the intersection of the addendum circle D with the rolling circle R_2) where contact commences, the driving tooth KA_2, *approaching* P, reaches that point when K and L coincide, the arc A_2P being called the **path of approach.** Obviously, the arcs PE, PF, and PG are the corresponding distances rolled, therefore these arcs are equal, the difference in the lengths of EF and EG being the distance that one tooth *slides* over the other in the same time.

It is an essential condition that one pair of teeth must not go out of gear before another pair comes into gear, and this condition is satisfied if the *pitch does not exceed either the arc* FPL *or* GPK, which are called **the arcs of action.** *These arcs are generally about* $1\frac{1}{2}$ *to* 2 *times the pitch.*

209b. Obliquity of Action, etc.—In all properly formed teeth in gear *the normal at the point of contact of two teeth always passes through the pitch point*[3] P (as EP does through P in Fig. 488); and neglecting the friction[4] of the teeth, the pressure Q between them is in the direction of this normal, which makes an angle with the tangent at P called **the angle of obliquity of action.**[5] Usually, with cycloidal teeth this angle has a maximum value of 30°. It is more clearly shown in Fig. 488A, where QP is the position of the normal, when contact begins, the angle θ decreasing until the point of contact reaches P (the pitch point), when it vanishes,[6] as the normal to the curves at that point coincides with the common tangent to the pitch lines. Fig. 488A also shows the direct way of determining the **path of contact**, EPQ, the points E and Q being the intersections of the rolling circles R and R_2, with the addendum circles D_2 and D respectively.

[1] Refer to the author's "Machine Design, etc.," p. 297.

[2] **Addendum**, the name given to the circle of the tips of the teeth.

[3] The condition which ensures the constancy of the velocity ratio is that *the common normal to two teeth at the point of contact must always pass through the pitch point.* Refer to the author's "Elements of Geometrical Drawing," Prob. 201.

[4] In cases where the friction must be reduced to a minimum, such as in delicate mechanisms, clockwork, etc., the expedient of shaping the wheels so that the driven teeth have no faces and the driving ones no flanks is often employed. Contact is then entirely confined to the *period of recess. The arcs of recess being then at least equal to the pitch.*

[5] If this be less than the friction angle, that is, the angle whose tangent is the coefficient of friction μ, there will be no pressure on the bearings. For rough cast iron $\mu = 0{\cdot}2$, corresponding to a friction angle $= 11{\cdot}3°$.

[6] This is the position of *direct thrust*, the normal having no components in the direction of the bearings of the shafts.

209c. How Form influences Durability.—*The longer the path of contact the larger the number of teeth that may be in gear at once*, and an examination of Fig. 488A will make clear that the path of contact (and therefore the working arc on the pitch lines) may be increased either by increasing the size of the rolling circles or of the addendum circles (and therefore the length of teeth), but we have seen that there are practical limits to the former increase, and the latter of course means weaker teeth, other things

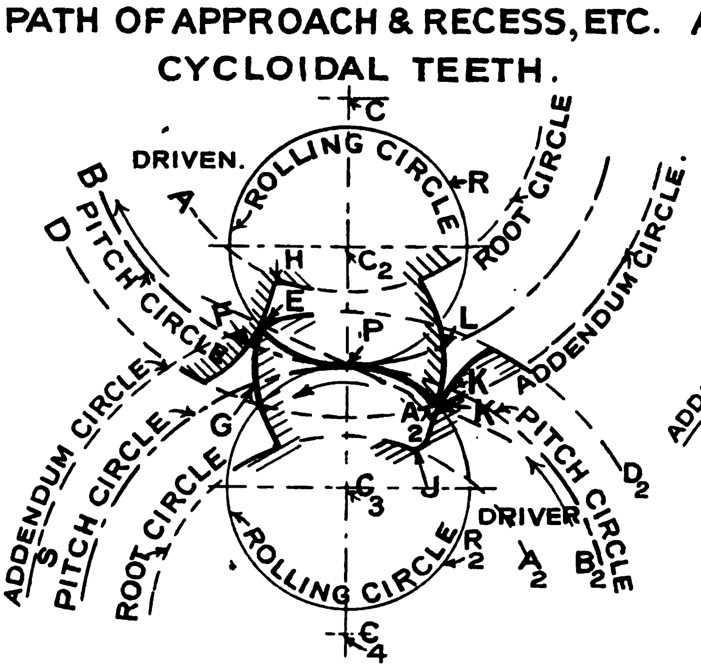


Fig. 488.

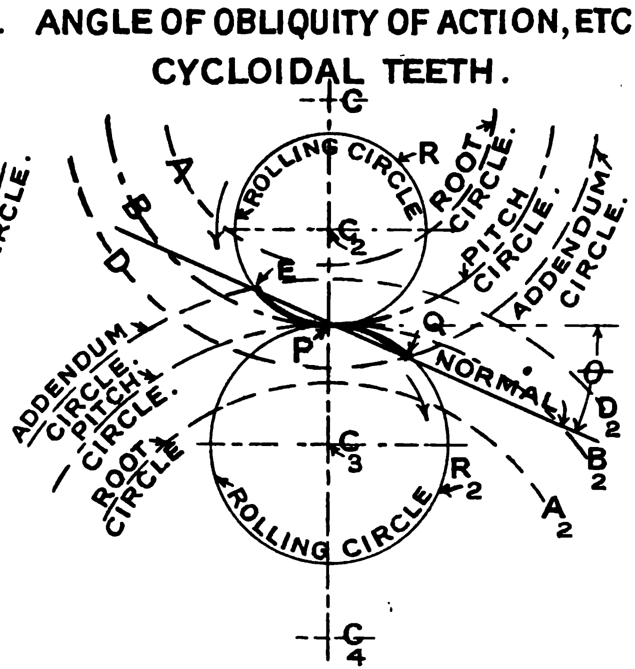


Fig. 488A.

being the same. Conversely, the general effect of reducing the size of rolling circles is (*a*) to decrease the arc or number of teeth in gear at once, and thereby *decrease* the power the wheel can transmit, also to increase the *obliquity of action* for a given length of tooth; (*b*) to increase the thickness of the teeth at the root, and thereby increase the power the wheel can transmit, by increasing the strength of the teeth; (*c*) to increase the wearing surface, and therefore the durability of the wheel. It will thus be seen that the best size rolling circle in any given case is in the nature of a compromise, and the relative value of each of these factors can only

be determined by trial in each case. But for trains of wheels such as are used for screw-cutting lathes and machine tools, the smallest wheel usually has 20 teeth, therefore the rolling circle for all the wheels may have a diameter equal to the radius of the 20-teeth one. Whilst for rough crane work the smallest pinions sometimes have a minimum of 11 teeth; on the other hand, the more important wheels used for transmission of power should never have less than 24 teeth, which corresponds to the line of contact (or obliquity) making an angle of 15° with the common tangent to the pitch lines[1] (or circles).

210. Proportions of Various Teeth, etc.—In Table 8 we have given the various proportions of wheel teeth in ordinary use. It will be noticed that in some respects they vary between pretty wide limits, so it will be as well to explain why this is so. Commencing with the clearance, obviously, a wheel cast from a wooden pattern, which has perhaps been stored in a damp place and has warped out of shape, and in the mould has, by irregular ramming, been still further distorted, will require more *clearance* both at the top and bottom and sides of its teeth[2] than one whose teeth have been shaped or machined accurately to form and size. Thus we have (in the second column of the table) *a clearance of* $0{\cdot}55p - 0{\cdot}45p = 0{\cdot}1p$ *in the ordinary wheels made from wood patterns*, which are almost of the proportions Fairbairn adopted (third column) in his extensive practice, whilst we see that **machine-moulded wheels** (fourth column) have only $0{\cdot}52p - 0{\cdot}48p = 0{\cdot}04p$ clearance, and that **machine-cut** wheels have practically no clearance[3] (eighth column); but of course these are only used under ideal conditions, where it is possible to make a pair of wheels

TABLE 8.—PROPORTIONS OF VARIOUS TEETH.

Parts of teeth. Refer to Fig. 481.	Common pattern-moulded wheels.	Fairbairn's proportions.	Machine-moulded wheels, say	Adcock's proportions.	Mortise wheels.	Mortise bevel wheels.	Machine-cut wheels (Browne and Sharpe).	
Pitch of teeth.	p	p	p	p	p	p	p	Diametral pitch p_d
Height above pitch line F .	$0{\cdot}33p$	$0{\cdot}35p$	$0{\cdot}3p$	$0{\cdot}2p$	$0{\cdot}25p$	$0{\cdot}25p$ to $0{\cdot}3p$	$0{\cdot}318p$	P_d
Depth below pitch line R .	$0{\cdot}42p$	$0{\cdot}40p$	$0{\cdot}4p$	$0{\cdot}3p$	$0{\cdot}3p$	$0{\cdot}3p$ to $0{\cdot}35p$	$0{\cdot}368p$	$1{\cdot}157p_d$
Thickness of teeth t . . .	$0{\cdot}45p$	$0{\cdot}45p$	$0{\cdot}48p$	$0{\cdot}48p$	cog, $0{\cdot}6p$; iron teeth, $0{\cdot}4p$	cog, $0{\cdot}6p$; iron teeth, $0{\cdot}4p$	$0{\cdot}5p$	$1{\cdot}571p_d$
Width of spaces S . . .	$0{\cdot}55p$	$0{\cdot}55p$	$0{\cdot}52p$	$0{\cdot}52p$	$0{\cdot}4p$	$0{\cdot}4p$	$0{\cdot}5p$	$1{\cdot}571p_d$
Total length l	$0{\cdot}75p$	$0{\cdot}75p$	$0{\cdot}7p$	$0{\cdot}5p$	$0{\cdot}55p$	$0{\cdot}55p$ to $0{\cdot}65p$	$0{\cdot}686p$	$2{\cdot}157p_d$
Width of teeth b	$2p$ to $3p$	Radius of fillet $0{\cdot}05p$	$2p$ to $3p$	$2p$ to $3p$	$2p$ to $3p$			

which shall always be in contact on both the working faces and the backs. But it more often happens that the teeth have from $\frac{1}{64}''$ to $\frac{1}{32}''$ clearance, according to size, etc. When the clearance is small, even with well-formed teeth there is often liability to

[1] Refer to the author's "Machine Design, etc.," p. 297.
[2] Refer to Plate No. 24, author's "Elements of Machine Construction and Drawing."
[3] Good *machine-cut* wheels can now be had at prices very little above those for wheels with ordinary *cast teeth*, and the time saved in fitting up soon repays the extra cost.

damage by the cross bearing of teeth due to wear of bearings or settlement, or by small objects falling into the wheels when at work. As to the **length of the teeth**, we have seen that the most suitable is a matter for judgment and experience. It is true that long teeth increase the arc of contact[1] (Art. 209a), and are indeed generally required to always remain in working contact at least in one place, when the pinions are small. But there can be little doubt that in mill-gearing, and similar cases where there are no small pinions, the length of the tooth can be reduced with a proportional increase of strength; indeed, the trend of modern practice has been for some years in this direction, particularly in the Lancashire district, and the length of half the pitch recommended by **Adcock**[2] has been proved by experience to be a perfectly satisfactory one, particularly for uncut cast gears. But the appearance of these short teeth to the uneducated eye militates against their more general use. The table shows (also Fig. 489) that Adcock made the *face length* $0{\cdot}2p$, and the *root length* $0{\cdot}3p$, which allows *bottom clearance* enough ($0{\cdot}1p$) for good fillets[3] at the root, as shown in the figure.

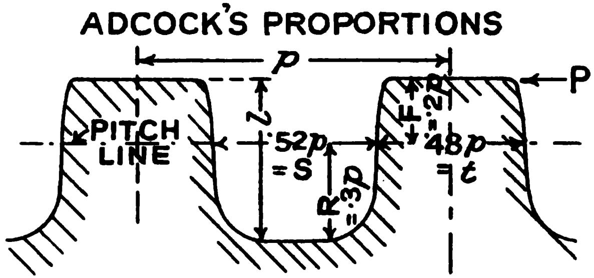


FIG. 489.

TABLE 8A.—DIMENSIONS OF MACHINE-CUT WHEELS (BROWNE AND SHARPE).

Pitch number. $1 \div P_d$	Diametral pitch. $P_d = \frac{\text{No. of teeth}}{\text{Diam. pitch circle}}$.	Circular pitch p.	Thickness of teeth $= 0{\cdot}5\,p$.	Height above pitch line $= P_d$.	Depth below pitch line.	Total length of teeth.	Pitch number. $1 \div P_d$	Diametral pitch. $P_d = \frac{\text{No. of teeth}}{\text{Diam. pitch circle}}$.	Circular pitch p.	Thickness of teeth $= 0{\cdot}5p$.	Height above pitch line $= P_d$.	Depth below pitch line.	Total length of teeth.
$\frac{1}{2}$	2	6·283	3·142	2·000	2·314	4·314	4	0·25	0·785	0·393	0·250	0·289	0·539
$\frac{3}{4}$	1·333	4·189	2·094	1·333	1·543	2·876	5	0·2	0·628	0·314	0·200	0·231	0·431
1	1	3·142	1·571	1·000	1·157	2·157	6	0·167	0·524	0·262	0·167	0·193	0·360
$1\frac{1}{4}$	0·8	2·513	1·257	0·800	0·926	1·726	7	0·143	0·449	0·224	0·143	0·165	0·308
$1\frac{1}{2}$	0·667	2·094	1·047	0·667	0·771	1·438	8	0·125	0·393	0·196	0·125	0·145	0·270
$1\frac{3}{4}$	0·571	1·795	0·898	0·571	0·661	1·233	9	0·111	0·349	0·176	0·111	0·129	0·240
2	0·5	1·571	0·785	0·500	0·578	1·078	10	0·100	0·314	0·157	0·100	0·116	0 216
$2\frac{1}{4}$	0·444	1·396	0·698	0·444	0·514	0·959	12	0·083	0·262	0·131	0·083	0·096	0·180
$2\frac{1}{2}$	0·4	1·257	0·628	0·400	0·463	0·863	14	0·066	0·224	0·112	0·071	0·083	0·154
$2\frac{3}{4}$	0·364	1·142	0·571	0·364	0·421	0·784	16	0·062	0·196	0·098	0·063	0·072	0·135
3	0·333	1·047	0·524	0·333	0·386	0·719	18	0·055	0·175	0·087	0·056	0·064	0·120
$3\frac{1}{2}$	0·286	0·898	0·449	0·286	0·331	0·616	20	0·050	0·157	0·079	0·050	0·058	0·108

[1] Refer to the author's "Machine Design, etc.," p. 299, for arc of contact, etc., of involute teeth.

[2] The *Engineer* of September 17th, 1869, also M. Longridge, has shown that even shorter teeth, from 0·35 to $0{\cdot}4p$, are better calculated to resist wear and tear. This matter has also been ably treated by Prof. Archibald Sharp, C.E., B.SC., WH. S.C., in his paper, "A New Method of Designing Wheel Teeth," *Proc. Inst. C.E.*, vol. cxiii. p. 241.

[3] This materially increases the strength of the tooth, particularly when subjected to shocks. In cases where it is not considered necessary to use an ample fillet, the root length of $2p$ may be correspondingly reduced. The table shows that Fairbairn in his extensive practice used a fillet of $0{\cdot}05p$ radius.

211. Gee's Buttress Teeth.[1]—In cases where a pair of wheels run always in the same direction the teeth may be strengthened by making their backs in the form of a buttress, as shown in Fig. 490, the driving faces of the teeth being of the usual form. In this way it is claimed that they can be made 35 per cent. stronger than ordinary teeth. The back faces may be described by smaller rolling circles or by involutes of considerable obliquity,[2] but whatever curve is used for them, there is an obvious increase of obliquity which makes this form of gearing quite unsuitable for use in cases where **back lash**[3] is likely to occur, as severe stresses upon the teeth, rims, and journals are caused by the wedging action of the back of the teeth.

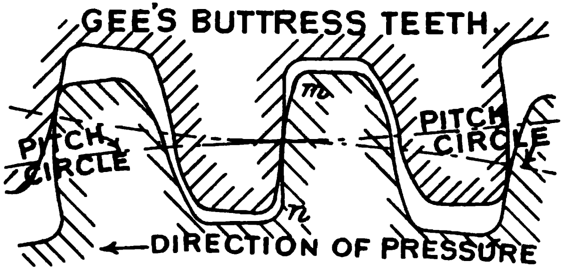


Fig. 490.

212. Knuckle Gearing.—A very strong but imperfect form of teeth *known as knuckle gearing, or* **Hollows and Rounds**, is shown in Fig. 491. It is sometimes used for rough crane work and other slow-moving machinery exposed to much rough treatment; and the teeth are formed by circles struck alternately within and without the pitch circles. As might be expected, the velocity ratio is variable, as the teeth come into and go out of contact.

213. Breadth of the Teeth.—When a tooth is engaged with its fellow, and is transmitting power, we have in some positions of the teeth in relation to one another *approximately* **line contact**, and therefore there is a *limit to the allowable pressure* per inch of breadth B (Fig. 492) on the face of the teeth apart from the strength of the teeth, which experience has proved should not be much exceeded. Fairbairn agreed with Tredgold's opinion and fixed this at 400 lbs. per inch of breadth for cast iron, and in ordinary cases this might well be taken as the limit.[4] Obviously the condition of loading of a tooth is that of a cantilever supporting a load at its free end, and therefore its strength is directly proportional to its breadth, while the pitch and form remain constant. But, should the axis of the shaft to which one of the wheels is fixed get out of its normal position, due to wear of the bearings at one end or to any settlement, it may happen that the load instead of being distributed over the whole width of the tooth is supported at one of the corners; or it may happen in a case where the clearance between the teeth was very small, causing them to bear on opposite corners, with a straining action enough to cause failure. Of course, other things being the same, the shorter the shaft the more serious this effect; for these reasons, it is only when there is a great probability of maintaining contact across the teeth that the *usual arbitrary breadth of* $2\frac{1}{2}$ *times the pitch* may be exceeded to the extent of 3 to $3\frac{1}{2}$ times the pitch. Box was of opinion that the breadth of $2\frac{1}{2}$ times the pitch makes the breadth of the teeth for a wheel of small pitch too

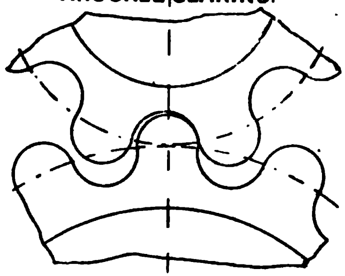


Fig. 491.

[1] Refer to the *Engineer and Machinist's Assistant*. These teeth were first suggested by Willis, in 1838. See Willis's "Mechanism," 2nd edit. p. 142.
[2] If all the curves be involutes, a large base-circle for the working sides *mn* is required, Fig. 490, and a small base-circle for the opposite sides.
[3] Refer to Art. 200, footnote.
[4] Although this 400 lbs. per inch of breadth should not be exceeded in a general way, **uncut** cast-iron teeth of very good design, well lubricated, have satisfactorily run with a load of 800 lbs. per inch.

broad, and one of large pitch too narrow, and recommended that the following formula should be used to fix the breadth, namely, **breadth of teeth** $B = p^2 \times 1{\cdot}8 \div \sqrt{p}$.

This gives for 1″ pitch, $B = \dfrac{1^2 \times 1{\cdot}8}{\sqrt{1}} = 1{\cdot}8''$; and for a 4″ pitch $B = \dfrac{4^2 \times 1{\cdot}8}{\sqrt{4}} = 14{\cdot}4''$.

214. Rims of Toothed Wheels.—Figs. 492 to 495 show the sections of toothed wheel rims in general use; the unit in each case is the pitch. Usually *the thickness of the rim is made equal to that of the teeth,* therefore $0{\cdot}48p$ has been assumed in these cases. For light work the section Fig. 492 is most suitable, and Fig. 495 shows a useful section generally used in heavy machine-moulded wheels.

RIMS OF TOOTHED WHEELS.

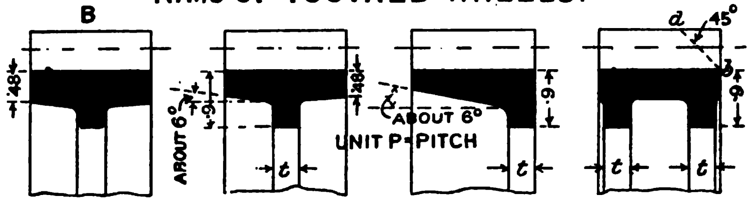


FIG. 492. FIG. 493. FIG. 494. FIG. 495.

215. Shrouding or Flanging of Wheel Teeth.—The strength of the teeth of wheels can be considerably increased by extending the width of the rim and carrying it outwards from the shaft, as shown in Figs. 496 to 498, the object being to reduce the *effective* length of the teeth as a cantilever, and thereby increase the breaking strength. Obviously, the amount of this increase will depend upon the form of the tooth to which it is applied, and the arrangement of the shrouding. In the case of a pinion gearing with a large wheel or rack, there is a great inequality of strength, the tooth of the pinion being much thinner at the root than that of the wheel or rack, and therefore it is much weaker. In many cases the one is only 0·7 the thickness of the other, and therefore has only half its strength;[1] but it can be shown that when this is so, by shrouding the pinion up to its pitch line, as in Fig. 497, the

SHROUDING OF WHEEL TEETH.

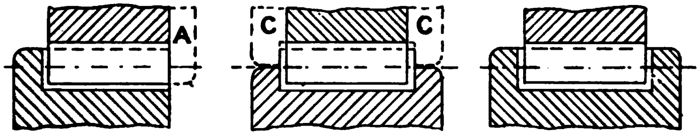


FIG. 496.—Single shrouding. FIG. 497.—Double shrouding to pitch line. FIG. 498.—Double shrouding. Whole length.

PROBABLE FRACTURES.

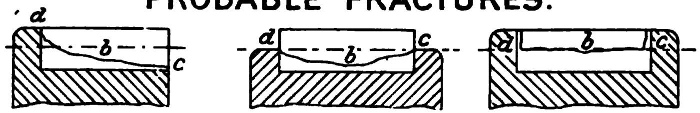


FIG. 499. FIG. 500. FIG. 501.

teeth have about the same strength. Further, as the teeth of the pinion are more often in contact than those of the wheel, they sooner become reduced in thickness by wear, and this should be borne in mind. The teeth of some large wheels are broader at the root than at the pitch line, and in form sensibly **approximate to a parabola**; when this is so it can be shown that they are practically equal in strength throughout their length, and the shrouding would be useless if the opposite teeth were

[1] The strength varies as the square of the thickness.

of the same material; but such wheels are sometimes shrouded if they gear with a pinion of stronger material, a cast-iron wheel and steel pinion, for instance, the wear being greatest in the former. And occasionally they are shrouded for appearance' sake only.

Another consideration which influences the designer is, that it may be more convenient to replace a pinion than a wheel, so that when the wheels may be subjected to unavoidable shocks, they sometimes shroud both wheel and pinion up to the pitch line,[1] as shown dotted in Fig. 497, or even shroud the wheel and leave the pinion plain,[2] as in Fig. 498. Of course, when teeth are shrouded right up to their points, as in this figure, failure must occur by shearing, probably along a line, *dbc*, Fig. 501, near the pitch line;[3] shrouding in this way about doubles the strength of the wheel, but, needless to say, only one of a pair can be made in this way. It is sometimes only convenient to shroud one side of the pinion, and gear it with a plain wheel or a wheel shrouded on the opposite side, as shown dotted at A, Fig. 496.

216. Bevel Wheels.[4]—We have seen, Art. 200, that in every pair of bevel wheels we have two imaginary cones (*or pitch surface*), Figs. 502, 503, and 504, rolling on one another with a common vertex *a*, which is the intersection of the axes of the two shafts. When the wheels are the same size and the shafts are at right angles, as in Fig. 502, we have what are called **mitre wheels.** Fig. 503 shows two unequal *bevel wheels* with shafts at right angles, and Fig. 504 a case where the shafts are not at right angles, but intersect at an obtuse angle θ.[5] The pitch point *p* is in the *common generator ap* of the cones, in each case. Obviously, when the angle between the shafts, the velocity ratio, and the diameter of the pitch circle of one of the wheels are given, the pitch cones can be easily set out. Figs. 505 and 506 are views of a mitre bevel wheel, and the few following hints bearing on the drawings of this wheel may be of interest to the young student.

217. Drawing Example. Cast-iron Mitre Bevel Wheel.—To draw the two views shown in Figs. 505 and 506, which are fully dimensioned, first draw the axis AL, and from any point A, set off the pitch line AP inclined 45° to it; a parallel to AL, and distant from it the radius of the pitch circle, will cut this line in P, the pitch point; through P draw DE at right angles to AP, and set off E and D the root and point of the tooth from P on this line. The length of the tooth may be now set off from P, on line PA, and the thickness of the rim EG from G, and finish these parts by lines from the vertex A, as shown, making the end of the tooth at F at right angles to the pitch line AP. From K, set off the length of the boss for L, and HL for the thickness of the web part. The centre N can then be found by bisecting the angles at H and L. The vertex B of a cone whose side is DG produced till it cuts the axis AB, can now be found and the arcs DS, PQ, and ET, forming part of the development of the back cone, described, and upon these the true shape of the teeth is set out, as shown. The circular arcs, of radii $2\frac{3}{4}''$ and $1\frac{3}{8}''$ forming the teeth, give very close approximations[6] to the true curve, and the drawings can now be easily finished

[1] Wheels gearing together that do not much differ in diameter may for ordinary cases be shrouded up to the pitch lines.

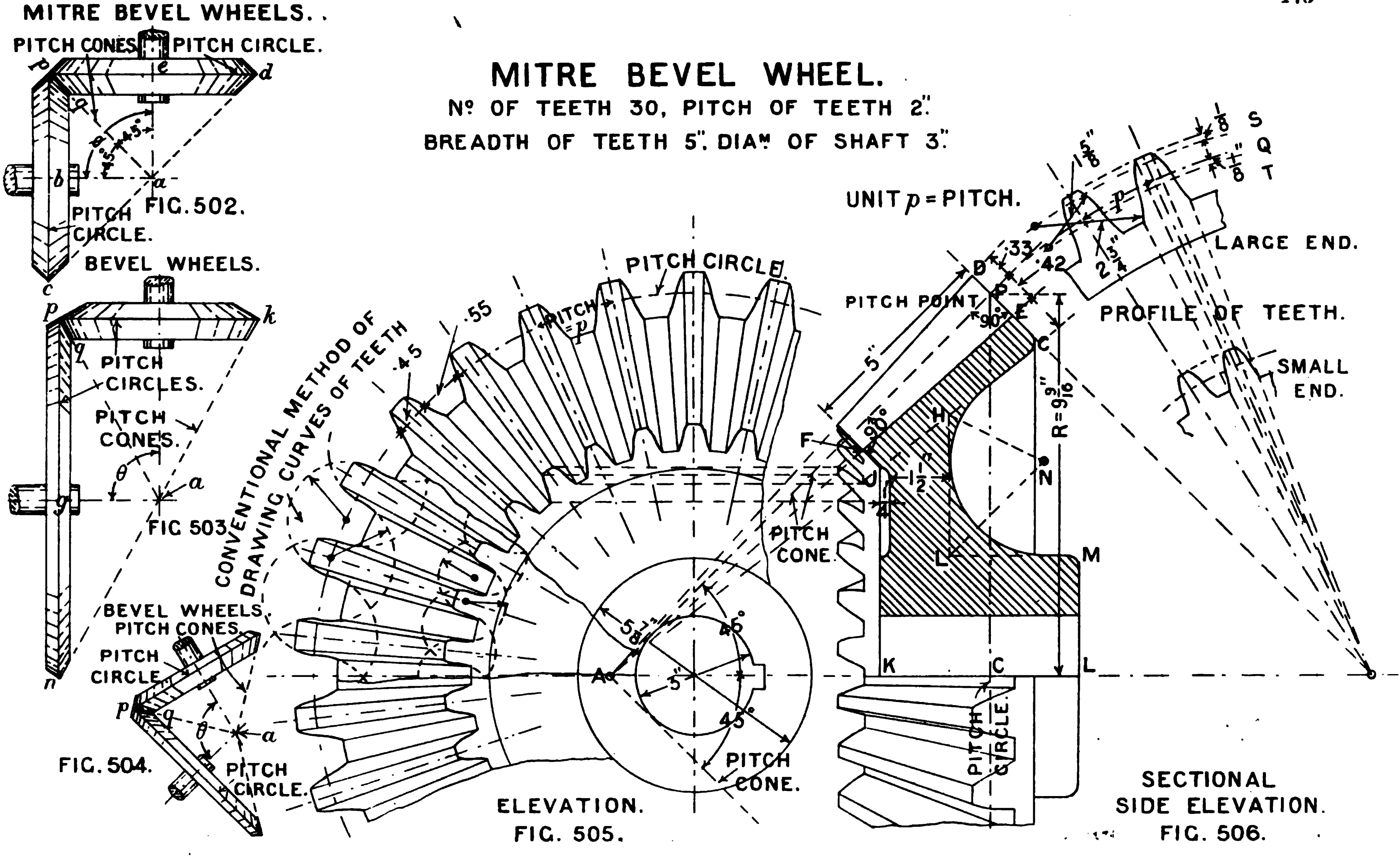
[2] Other expedients are making the teeth of different thickness, and the use of frictional slipping devices.

[3] Figs. 499 and 500 show at *dbc* the probable lines of fracture for the other methods of shrouding.

[4] Formerly, in large machine shops the long lengths of line shafting, arranged parallel to one another at short intervals in the length of the shop, were often driven from a main shaft which ran along the side of the shop, by means of bevel wheels; but since electric motors have been so largely used, each to drive a single line of shafting, bevel wheels for such purposes have been discarded.

[5] We have seen that when two shafts are not quite in the same straight line, and one drives the other through a Hooke's joint, there may be an ununiformity in the motion, and perhaps excessive wear. But bevel wheels can sometimes be used instead, making a much better mechanical job.

[6] Each arc passes through three points in the actual cycloidal curve, this curve being first geometrically set out (or by the use of the odontograph or templets), as previously described. See author's "Elements of Machine Construction and Drawing," Plate No. 24.


MITRE BEVEL WHEELS.
PITCH CONES.
PITCH CIRCLE.
PITCH CIRCLE.
FIG. 502.
BEVEL WHEELS.
PITCH CIRCLES.
PITCH CONES.
FIG 503
BEVEL WHEELS PITCH CONES
PITCH CIRCLE
PITCH CIRCLE.
FIG. 504.
MITRE BEVEL WHEEL.
Nº OF TEETH 30, PITCH OF TEETH 2".
BREADTH OF TEETH 5". DIAM. OF SHAFT 3".
UNIT p = PITCH.
CONVENTIONAL METHOD OF DRAWING CURVES OF TEETH
PITCH CIRCLE.
PITCH = p
PITCH CONE.
PITCH CONE.
ELEVATION.
FIG. 505.
PITCH POINT
LARGE END.
PROFILE OF TEETH.
SMALL END.
PITCH CIRCLE.
R = 9 9/16"
SECTIONAL SIDE ELEVATION.
FIG. 506.

without further help, but it should be explained that as the teeth curves on the elevation are foreshortened, they are drawn in the conventional way *shown* by the dotted arcs.

218. Strength of Wheel Teeth.—Having considered how to obtain the best form for the teeth of wheels, we may now proceed to see how their *size* for any given case may be determined, and this is to a large extent a question of strength, which depends upon (*a*) the strength of the material, (*b*) the forces which act on the teeth due to the power transmitted, (*c*) the way in which the teeth resist fracture under the action of the load which comes upon them. First, with regard to (*a*) the material, **cast iron** on account of its cheapness and because it may be readily cast in any form, is used for ordinary wheels; second, (*b*) with small pinions, such as are used in rough crane-work, only one tooth of each wheel can be relied upon to engage at once; indeed, for the matter of that, owing to slight inaccuracies, and the possibility of the presence of dirt between the teeth, *it is probably never absolutely safe to rely upon there being more than* **one pair of teeth in actual working contact at once,**[1] whatever the size of the wheels. So, assuming this to be the case, we have the force P (acting in the pitch circle of the driving wheel) acting as a load on the end of a tooth (as in Fig. 488A), and we may regard the case as one of a cantilever loaded at its end. And, lastly, (*c*) the tooth, if in contact throughout its breadth with its fellow, may break at its root, and the full strength of the tooth is available, or, should the load P on a tooth act at one corner only, owing to faulty construction or erection, or to settlement or wear of a bearing, then the tooth may break along a line *ab*, Fig. 495, making about 45° with its root.[2] We may now examine the strength of a tooth, assuming what we may call *the* **normal case** *of one pair of teeth in full contact,* taking the proportions we have in the Table (8) for *machine-moulded* wheels, as being fairly representative of good modern practice,[3] we have the length of the cantilever $l = 0{\cdot}7p$ and its breadth b say $2{\cdot}5p$, whilst if we **allow for 25 per cent. wear** of the teeth,[4] the thickness t becomes $0{\cdot}48p - 0{\cdot}12p = 0{\cdot}36p$. Then, equating greatest bending moment to the moment of resistance to bending, we get $Pl = Zf$, where Z, the modulus of the section, is $\frac{bt^2}{6}$, and f, the safe skin stress

$$= \frac{\text{ultimate skin stress}}{\text{factor of safety}} = (\text{say for this example}^{5})\ \frac{36{,}000}{10} = 3600 \text{ lbs. per sq. inch.}$$

$\therefore$ **The safe load P on one pair of teeth** (of 1″ pitch) $= \dfrac{2{\cdot}5 \times 0{\cdot}36^2 \times 3600}{6 \times 0{\cdot}7} = 278$ lbs. . . . . (34)

And for Two pairs of teeth in contact $\quad P = 2 \times 278 = 556$lbs. . . . . (35)

[1] Some writers get over this difficulty by assuming that each tooth takes about two-thirds the full load, but surely it must *be either one tooth or two teeth in contact*, and to be on the safe side we shall assume the former. The student will readily be able to make the proper allowance for any other assumption.

[2] It can be shown that its strength to resist failure in this way is about equal to that of the tooth with a breadth of about $1{\cdot}4p$, or approximately twice the length of the teeth. By some, a greater breadth than this is not reckoned to add to the transverse resistance of the tooth; but it is necessary for durability, so that the maximum breadth that can be relied upon under all conditions is twice the length.

[3] These proportions are frequently used both for *cut* and *uncut* teeth. Wheel teeth are now made of better form and proportions than they were years ago, with an improvement in the uniformity of loading and strength.

[4] The allowance recommended by Tredgold, and adopted by Fairbairn and Unwin. See Fairbairn's "Millwork," vol. ii. p. 43.

[5] See Anderson's "Strength of Materials," p. 188. A cast-iron bar, 1″ long and 1″ square, loaded at its end as a cantilever, breaks with about 6000 lbs.

Then we may take, to find the equivalent skin stress f, $\quad 6000 = \dfrac{bt^2f}{6l}$, or $f = \dfrac{6000 \times 6}{1 \times 1^2} = 36{,}000$,

and with FS (factor of safety) = 10, we get $\quad f = \dfrac{36{,}000}{10} = 3600.$

The student should now be in a position, in any given case, to decide which of the above values would be applicable. Now, the **factor of safety** part of the equation requires some further consideration, for only those who can intelligently decide upon its allowable value in any given case can succeed in steering clear of the mistakes which can so easily be made, due to the apparent want of agreement that exists in the works of various authorities; so that when the failure of a wheel occurs it is not difficult to justify its proportions by reference to some accepted authority. And this is possible, because frequently the *conditions of running* that are assumed are not sufficiently defined, or are so involved that none but the expert can grasp their true significance and value in a given case. This can be better understood by an inspection of Table 9, which has been calculated from the equations 34 and 35, for the cases of *one pair and two pairs of teeth in contact* and for a range of value of the *factor of safety* used, which satisfies most conditions of running in practice.

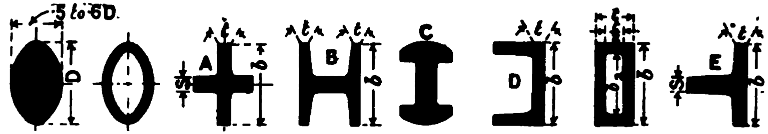
TABLE 9.—Safe Load P, for Cast-iron Teeth of 1″ Pitch and Breadth of 2·5 Times the Pitch. When the Pressure is distributed uniformly over the Breadth of the Teeth,[1] the Ultimate Skin Stress being taken at 36,000 lbs.

Kind of running.	No. of case.	One pair of teeth in gear at once, P =	Two pairs of teeth in gear at once, P =	Skin stress f in lbs.	(F.S.) Factor of safety used.
Without shocks	1	463	926	6000	6
Very slight shocks . . .	2	278	556	3600	10
Moderate shocks	3	139	278	1800	20
Excessive shocks	4	93	185	1200	30

It will be noticed that the above values are for breadth of $2\cdot5p$, but of course for **any other breadth** (when P is distributed uniformly over it) the value of P will be directly proportional to the breadth.

219. Arms of Wheels, their Shape and Strength, etc.—Only very small wheels (called plate wheels) are made with a disc connecting their rims to their naves. The number of arms that a given wheel should have is more or less arbitrary, their number increases with the size of the wheel, usually wheels under 4′ having four, from 4′ to 8′ six, and 8′ to 16′ eight. Pulleys, or quite light wheels, have arms of *elliptical* section, Fig. 507, and the *cross* section, Fig. 509, is largely used, whilst the **H** section, Fig. 510, is more often used for *heavy wheels; the tee section,* 514, *is commonly used for bevel wheels,* and the other

VARIOUS SECTIONS OF WHEEL ARMS.



Figs. 507. 508. 509. 510. 511. 512. 513. 514.

[1] Refer to author's "Machine Design, etc.," p. 310.

sections, Figs. 511 and 512, are more often adopted for built-up wheels; and when these wheels have *steel arms* the hollow sections, Figs. 508 and 513, are generally used. Obviously, when the arms, rim, and nave are cast in one piece, the arms are fixed at both ends, but *if the arms are in any way attached to the rim, they are usually* **treated for strength purposes as being fixed at the nave only**; indeed, for the matter of that, to make allowance for the contraction in cooling stresses, it is usual to assume that all arms are fixed in this way.

220. Naves or Bosses of Wheels.—We have, in Figs. 515 to 521, forms of the naves or bosses of wheels which correspond to some of the arms shown in Figs. 509 to 514, the letters A, B, etc., being common to both sets. When the wheels are large and heavy the initial stresses due to contraction in cooling may seriously reduce the strength of the nave; to avoid this they are made in two parts and bolted together. Figs. 522 and 523 show the nave part of such a wheel. Or the nave is sometimes slotted across between the arms in two or more places, according to the size of the wheel,[1] and iron plates are fitted to the openings, a ring of wrought iron R being shrunk on each side of the nave to bind the segments firmly together.

NAVES OR BOSSES OF WHEELS.

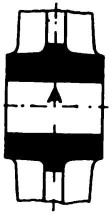


FIG. 515.

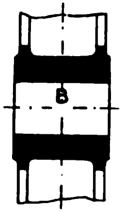


FIG. 516.

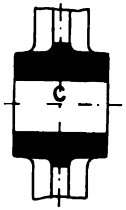


FIG. 517.

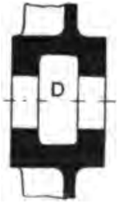


FIG. 518.

FIG. 519.

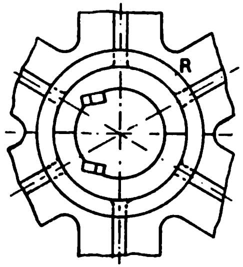


FIG. 520.

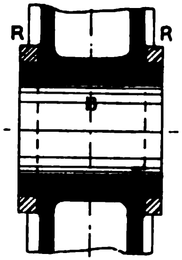


FIG. 521.

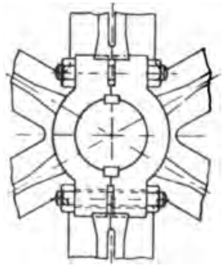


FIG. 522.

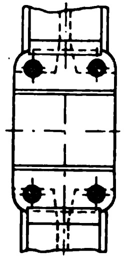


FIG. 523.

There is apparently no very satisfactory rule in general use for the **thickness T, and length of the nave of a wheel.** For spur wheels—

$$\textbf{Box's rule is } T = (p \times 7 \div 9) + (0{\cdot}125 \times D) \quad \ldots\ldots\ldots \quad (36)$$

where p = pitch of teeth in inches, and D = diameter of wheel in feet.

$$\textbf{Unwin's rule is } T = 0{\cdot}4\sqrt[3]{p^2R} + \tfrac{1}{2}''$$

where R is the radius of the pitch circle in inches.

A pretty general arbitrary practice (when a designer in a particular case has not experience to guide him) is to make them with a diameter at least twice that of the shaft at its smallest section (usually at the journal), or, if d be the diameter of this part, and the

[1] Refer to author's "Machine Design, etc.," p. 317.

shaft is increased in size at and near the wheel to resist bending, then T, the thickness of the nave, is made $\frac{1}{2}d$. For a parallel shaft the wheel seating is usually $1{\cdot}17d$ to $1{\cdot}18d$ in diameter, the **length of the nave** being from $1{\cdot}5d$ to $1{\cdot}75d$.

221. Rims of Wheels.—A few typical sections of wheel rims principally for fly wheels, are shown in Figs. 524 to 536. The rims, Figs. 530, 531, and 533 are for belt drives, whilst 529 is for a band saw wheel, the arms being *staggered* to allow for contraction

RIMS OF SOLID AND BUILT-UP FLY WHEELS.

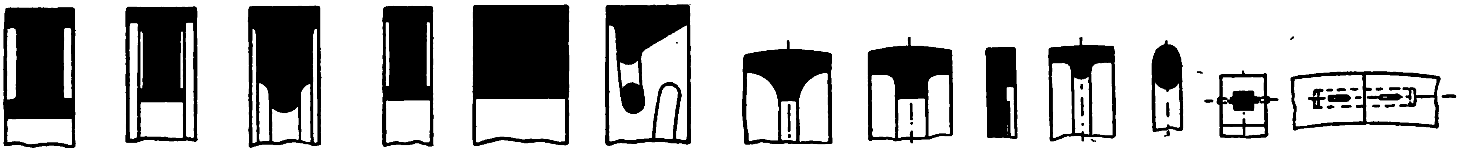


FIG. 524. FIG. 525. FIG. 526. FIG. 527. FIG. 528. FIG. 529. FIG. 530. FIG. 531. FIG. 532. FIG. 533. FIG. 534. FIGS. 535, 536.—Cottered dowel.

in cooling and to give lateral stiffness. The other sections are for fly wheels proper. Obviously, Figs. 528 and 534 are simple forms whose moments of inertia can be readily found. Fig. 532 is a section suitable for attaching arms in built-up wheels, and Figs. 535 and 536 are two views of a built-up rim with cottered joints.

222. Mortise Wheels.—When wooden teeth are mortised and fixed into rims of cast-iron wheels designed to receive them, as shown in Figs. 537 to 553, we have what are technically called *Mortise Wheels.* The wood commonly used for the cogs is **hornbeam**, which, owing to its *strength*[1] *and stringy toughness, is unsurpassed* by any other for the purpose, although **birch** is occasionally used for cheapness' sake. The taper part or tenor of the well-seasoned cog is driven tight[2] into the rim of the wheel, and usually secured by a pin of iron (drawn) wire, the hole-for which is so placed that when the pin is driven in it tends to draw the cog still further into the wheel. Fig 540 shows a cog with short tenon for fitting opposite an arm.[3] An alternative way of securing the cog is shown in Fig. 544, dovetailed wooden keys being driven between the projecting ends of the tenons. A variation in the form of the key (used on the Continent) is shown in Figs. 545 and 546, whilst in 547 we have a saw-cut in the end of the tenon, so that the end may be spread when the pin is driven. Fig. 551 shows how the arms are formed near the rim to allow of the cogs being fitted at these parts. It will be noticed that the cog A, Fig. 541, is symmetrical about a centre line through the tenon, whilst cog B (Fig. 542) is flush at one of its sides C, which means that the other, the *working side*, has more material available for wear. When the wheel is wide, two or more cogs are used to make up the breadth, as shown in Fig. 539. The usual proportions are shown on some of the figures. When cogs are used there is little or no clearance required between the cogs and iron teeth with which they gear, generally the cog is $0{\cdot}6p$ in thickness. Figs. 552 and 553 show sections of the rim of a **bevel mortise wheel**, Fig. 548 showing how

[1] A cantilever bar of this wood, 1″ square, 1″ long, breaks with about 1800 lbs. at its free end (about three-tenths the breaking weight of cast iron).

[2] After the rough cog has been fitted and driven into the rim, a line is scribed round the cog a uniform distance from the surface of the rim, and the cog knocked out, when it is shouldered with the wood chisel, so that when it is driven in again it can be driven right up to the shoulder, as in Figs. 537 to 543, or show a uniform clearance, as in Figs. 545 to 547. Refer to author's "Machine Design, etc.," p. 322.

[3] Refer also to Sheet 39, author's "Elements of Machine Construction and Drawing."

DETAILS OF MORTISE WHEELS.

UNIT = PITCH = p

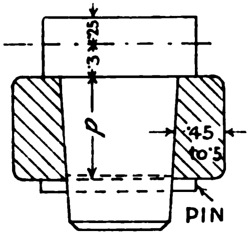


FIG. 537.

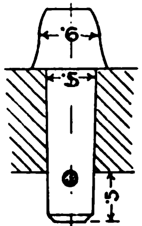


FIG. 538.

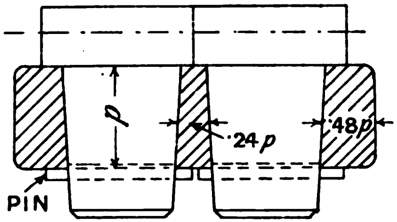


FIG. 539.

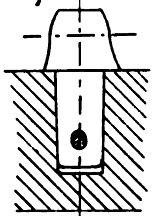


FIG. 540.

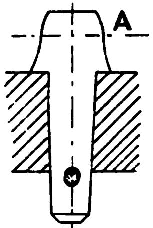


FIG. 541.

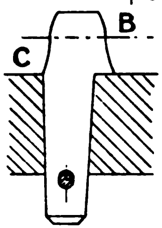


FIG. 542.

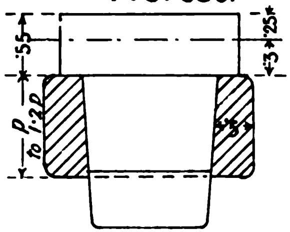


FIG. 543.

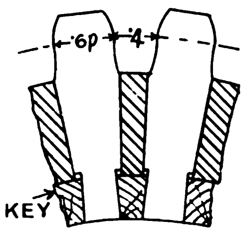


FIG. 544.

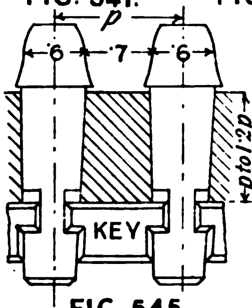


FIG. 545.

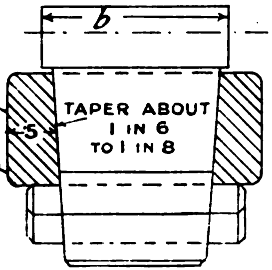


FIG. 546.

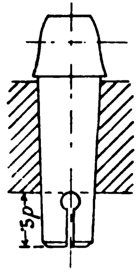


FIG. 547.

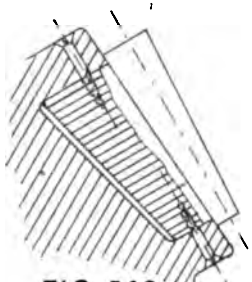


FIG. 548.

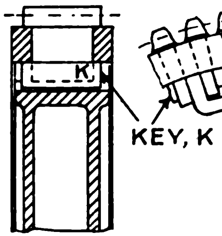


FIG. 550.

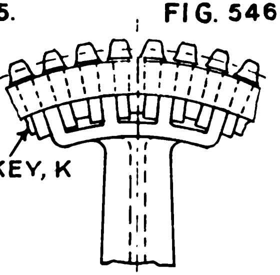


FIG 551.

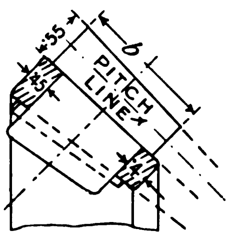


FIG. 552.

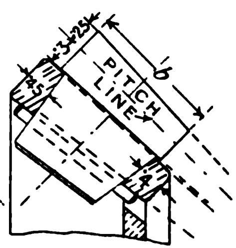


FIG. 553.

cogs near the arms are secured by screws.[1] *Mortise wheels are used with the object of introducing an* **elastic medium** *to reduce the effect of shocks due to any defect in the form of the teeth, and to* **reduce the noise** *that is commonly made when ordinary iron teeth are running together.* When the iron teeth of wheels that are to gear with mortise wheels are *machine moulded*, the teeth only, as a rule, require filing to clear them of the sand and make them smooth, but in cases where the wheels are cast from patterns, the teeth must be machined or *pitched and trimmed*, the rough surfaces being chipped true to their geometrical form, and filed smooth; the wooden teeth then easily wear to the exact form of the iron teeth they gear with, without their surfaces being destroyed by the rough surfaces of the casting. But the steady improvement in the construction of wheels with iron teeth, which has been taking place for some years (particularly since the introduction of improved **helical gearing**), has caused **mortise wheels** to be **comparatively little used**; indeed, *it is not often that their adoption is now really necessary*, mainly owing to the great improvements that have been made in the cutting of helical gears, a description of which is given in the author's "Machine Design, etc.," p. 326.

EXERCISES.

Designing, etc.

1. Set out the pitch circles of a pair of spur wheels, velocity ratio 3 and 2. The smaller wheel has thirty teeth, and the pitch is $1\frac{1}{4}''$. Be careful to give the exact distance between their axes. Scale $3'' = 1'$.

[1] Refer also to author's "Elements of Machine Construction and Drawing," Plate No. 40.

2. A toothed wheel has fifty teeth, whose diametral (or Manchester) pitch is No. 4. Set out its pitch circle and give the outside diameter of the teeth. Scale 3″ = 1′. What is the *circular* or true pitch of the teeth of this wheel?

3. The circular pitch of the teeth of a wheel is 22 millimetres, and the number of teeth 120. Set out the wheel pitch circle, and give the diameter of the pitch circle, and the outside diameter. If it gears with another wheel half its size, what is the difference between the axes? Scale $1\frac{1}{2}''$ = 1′.

4. The pitch circles of a pair of wheels in gear are 4″ and $2\frac{1}{4}''$ in diameter, and the rolling circle is $1\frac{1}{4}''$ diameter. Determine the arc of action, and the arcs of approach and recess of the teeth described by the rolling circle.

5. Referring to the previous exercise, measure the obliquity of action of the teeth, assuming that the faces of the teeth are $\frac{3}{16}''$ long on your drawing.

6. Assuming that a wheel with teeth of 1″ pitch, whose breadth is $2\frac{1}{2}p$, safely transmits 140 lbs. at the pitch line when subjected to moderate shocks. What should be the pitch of the teeth of a wheel to transmit 30 H P. under similar conditions at a speed of pitch line 1800 feet per minute? *Ans.* 1·98″. Say, pitch = 2″.

Drawing Exercise.

7. Set out the mitre bevel wheel shown in Figs. 505 and 506. Scale half full size.

Sketching Exercises.

8. Sketch two *cycloidal teeth*, roughly in good proportion, pitch = 4″, and mark on them the names and proportions of the various parts. You may make the length of the teeth 0·7 pitch.

9. Sketch a pair of Gee's *buttress teeth* in gear, and explain what advantage is claimed for this form of teeth. What is their principal disadvantage?

10. Show by sketches how the teeth of wheels are shrouded. What is the object of shrouding? Describe the kind of wheel which most requires shrouding.

11. Sketch four typical sections of wheel arms, and explain for what type of wheel you would use each one.

12. Show by sketches how the nave of a wheel is strengthened by shrinking on wrought-iron rings.

13. Make a sketch of two teeth of a mortise wheel in fairly good proportion, and show how the cogs are secured. Also show an alternative way of securing the cogs, and give the principal proportions. What is the object of using a mortise wheel?

14. Make sketches showing how cogs which come near the end of an arm in a mortise wheel are secured: (*a*) when the wheel is solid; (*b*) when the wheel is made in halves.

15. Show a section of the rim of a mortise bevel wheel, with a cog in position, and mark on the sketch the principal dimensions.

16. Show by sketches two ways of connecting by toothed wheels, shafts that cross one another but are not in the same plane.

17. Sketch a light fly wheel with *staggered* arms. What advantages are claimed for this arrangement of the arms?

18. Sketch four or five representative sections of *fly wheel rims*. What advantage can be claimed for a simple rectangular section?

CHAPTER XX

BELT GEARING

223. Fast and Loose Pulleys.—There are some interesting and important details of these that should receive attention. In Fig. 556 the shaft A of a machine projects from the frame bearing B, and is fitted with the fast and loose pulleys F and L, the former being driven on to the feather K, which fixes it to the shaft, and the latter (bushed with gun-metal) is kept in position by the washer W, which is held on to the end of the shaft by the hollow brass screw S, whose ball end is drilled and countersunk to admit oil for the bearing; an alternative fitting here being a Stauffer's grease-box. A cheaper but less efficient arrangement for lubricating is shown at *x*, Fig. 558, but the tendency of the oil is to flow out of the hole away from the journal due to centrifugal force, whilst, if it is admitted from the axis, as in Fig. 556, in flowing outwards it passes over the journal. Fig. 557 shows a part of the rim of the driving wheel, whose width must of course be at least equal to both that of F and L, the fast and loose pulleys, Fig. 556. Fig. 558 also shows how a screw D is sometimes used to keep the bush in position; a more usual way is to slightly countersink each end of the hole in the boss and to burr the bush over, the latter being in all cases a driving fit in the boss. The loose collar C, which keeps the pulley in position on the shaft, is fixed to the shaft by the set-screw E. When the loose pulley is not bushed, the boss is made much longer, as shown in Fig. 559. This figure also shows how set-screws T are sometimes used with feathers, to avoid driving the pulley on in fixing.

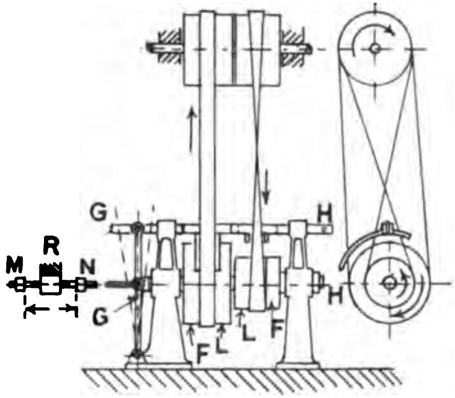
In cases where the belt is running on the loose pulley the best part of the time, it is advantageous to make this pulley somewhat smaller in diameter than the fast pulley[1] (Fig. 557A). This **relieves the tension** of the belt and the pressure on the journals when the belt is running idle. In this arrangement it will be seen that the bracket A has a sleeve S projecting from the boss B, upon which the loose pulley L works, the hole throughout the boss and sleeve being larger than the shaft, to clear it, so that the latter runs perfectly clear of the loose pulley when the belt is on the fast pulley F, which is **coned at its edge** C to allow the belt to easily mount the larger pulley. Of course in this case the belt, which is driven from the main shaft, is always running.

Figs. 554 and 555 are two views of a belt gear for slow forward and quick return motion, which should speak for themselves.

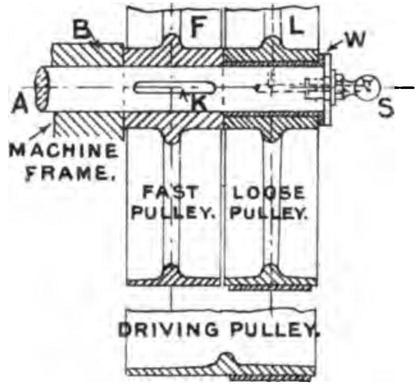
224. Rims of Pulleys or Riggers for Belting.—As **lightness** and **rigidity** are two essential features of these wheels, the rims are made as thin as practicable, the inside of the section tapering slightly, as shown in Fig. 566 (which gives suitable proportions), so that the pattern may be easily drawn from the mould, and a stiffener or strengthening rib S (Figs. 560 to 564) is generally used to assist in making them rigid. With wide pulleys the stiffeners are sometimes on the edges, as in Fig. 563, and when they have double sets of arms these are usually placed somewhat near the edges, as at S, Fig. 564; but the inner surface

[1] **Refer to author's "Machine Design, etc.," p. 375.**

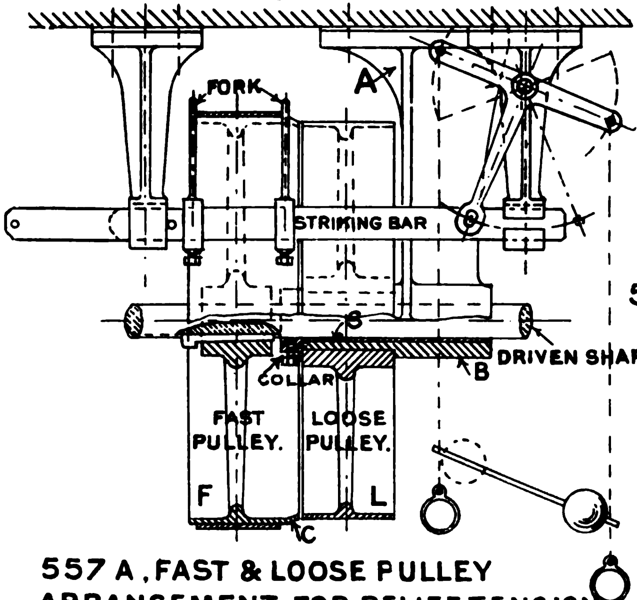
FAST AND LOOSE PULLEYS, ETC.



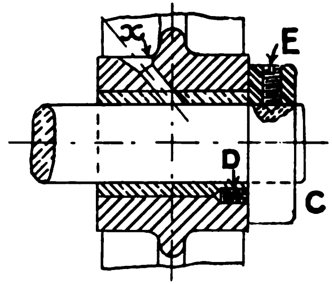
554 & 555. BELT GEAR FOR SLOW, FORWARD, & QUICK RETURN MOTON.



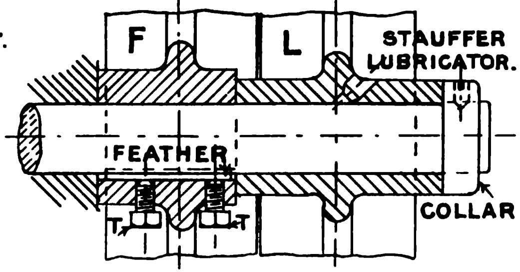
556 & 557, FAST & LOOSE PULLEYS FOR SHAFT OF A MACHINE.



557 A, FAST & LOOSE PULLEY ARRANGEMENT FOR RELIEF TENSION.

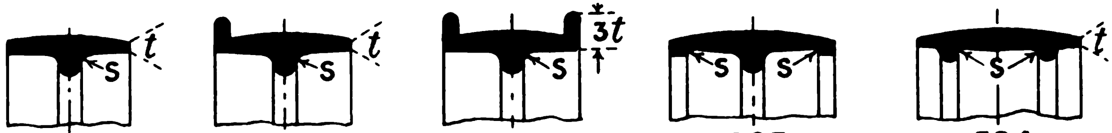


558, DETAIL OF BUSHING - LOOSE PULLEY



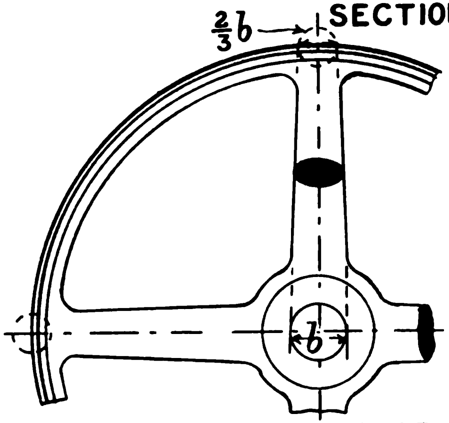
559, FAST & LOOSE PULLEY WITHOUT BUSHING.

SOLID BELT PULLEYS.

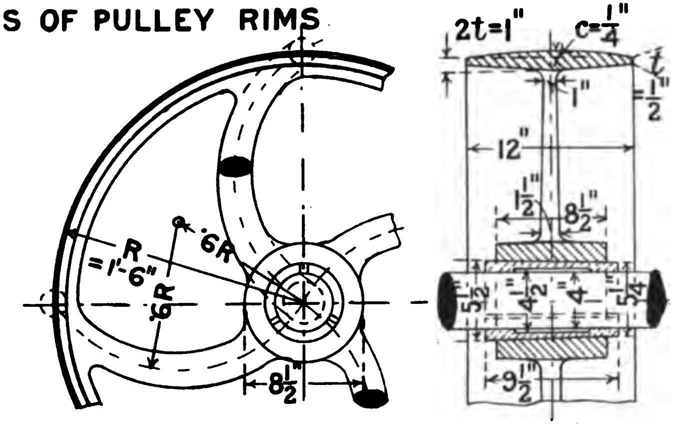


560. 561. 562. 563. 564.

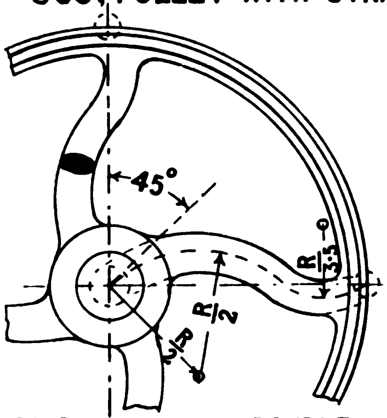
SECTIONS OF PULLEY RIMS



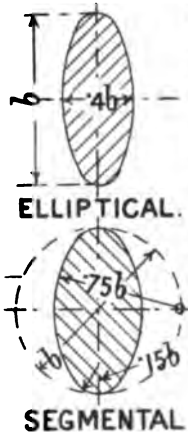
565, PULLEY WITH STRAIGHT ARMS.



566, PULLEY WITH CURVED ARMS.



567, PULLEY WITH DOUBLE CURVED ARMS.



ELLIPTICAL.

SEGMENTAL.

568, SECTIONS OF PULLEY ARMS.

of the rims should be turned as near the arms as possible to balance the wheel (the importance of this increasing with the square of the speed), so this should be a guide in designing them. *The* **thickness** (t) **of the rim at the edge** after turning may be about $t = 0{\cdot}6T + 0{\cdot}0003D$, for pulleys where B does not exceed $\frac{1}{4}D$ to $t = 0{\cdot}7T + 0{\cdot}0005D$ for wider pulleys, where T is the thickness of the belt, B is the breadth of face in inches, and D is the diameter of the pulley in inches. The width of belt pulleys is usually at least $\frac{1}{4}''$ greater than that of the belt, or, say, width of belt $= 0{\cdot}9B$.

225. Proportions of Pulley Arms.—The arms of pulleys are either *straight*, as in Fig. 565, *curved*, as in Fig. 566, or *double curved*, as in Fig. 567. The object of curving is to prevent fracture when the casting cools, but with straight arms, suitably proportioned and with proper precautions taken in cooling, there should be no trouble from contraction. This being so, they should be preferred, as they are lighter and stronger, and their patterns are less costly. The section of the arms tapers off from the boss to the rim in the way shown in Fig. 565. The usual practice is to make the breadth and thickness at the rim two-thirds the amounts at the boss. Fig. 566 shows how the curved arms may be set out. The **number of arms** is fixed in a somewhat arbitrary way, the usual practice being to have **four** for pulleys up to about 24″ diameter, **six** from about 2′ to 8′ or 9′, and **eight** for larger sizes, **ten** being sometimes used for **very large** wheels. But the breadth of the pulley should properly

be a factor in fixing the number of arms, and Professor Unwin gives the number of arms $N = \frac{BD}{150} + 3$, the breadth B and diameter D being in inches, and the nearest whole number being taken. The **shape of the rim section** is either elliptical or segmental, of the proportions shown in Fig. 568.

226. Split Pulleys.—Some shafts are bossed at their ends, and then, to avoid using *cone keys*, the pulleys are often cast in halves and bolted together in position without dismounting the shaft. Figs. 569 and 569B show two examples of this class of pulleys,[1] Fig. 569A showing how the joint is made when it occurs between arms. The bolt, or bolts, at the rim and boss should have a total net section equal to about 0·28 to 0·3 times the sectional area of the rim and boss respectively. In recent years there has been a great development in the use of **wrought-iron** and **steel pulleys**, particularly for high speeds and large diameters; owing to their lightness, and freedom from the initial strains due to cooling which exist in cast-iron pulleys; they have the additional advantage

SPLIT BELT PULLEYS.

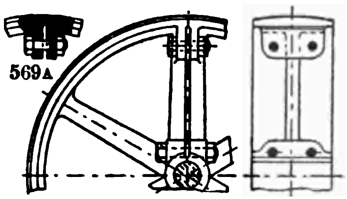


FIG. 569.—Cast-iron split pulley.

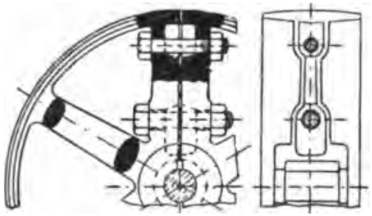


FIG. 569B.—Cast-iron split pulley.

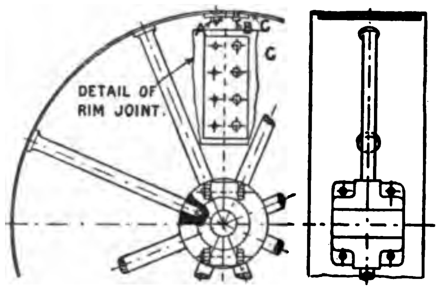


FIG. 569D.—Wrought-iron split pulley.

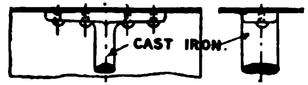
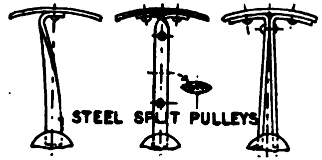


FIG. 569C.—Medart's split pulley.



Macbeth's. Universal. Mackie's.
FIG. 569E.

of not flying to pieces should they be overrun. Fig. 569D shows a pulley of this type; the boss is made of cast iron in two parts and the rim in one piece, the ends joined by a cover strip or lapping piece, placed under the rim and riveted to one part and bolted to the other. Both joints can be sprung open wide enough to receive the shaft. The bolt springs in all split pulleys are made to grasp the shaft tight enough to give a frictional drive without keys. Fig. 569C shows a **Medart's split pulley**, which has a wrought-iron rim, with the arms and boss of cast iron. Fig. 569E shows three well-known forms of steel split pulleys. The arms and rims of this type are made of steel, and the bosses or hubs are either cast iron or cast steel.

227. Thickness and Length of Bosses or Naves of Pulleys.—There seems to be no well-established rule for determining the thickness of the boss (sometimes called the *eye* or *hub*) or central part of a pulley, as we remarked in Art. 220 in referring to spur wheels.

[1] In quite small pulleys bolts at the boss only are required.

It seems obvious that it should either vary with both the size of shaft and diameter of pulley, or with both the breadth and diameter of the pulley. Unwin[1] favours the latter, and says the **thickness** t **of the boss** *may be* as follows:—

$$t = 0{\cdot}14\sqrt[3]{\mathrm{BD}} + \tfrac{1}{4}'' \text{ for single belt} \qquad (37)$$

$$t = 0{\cdot}18\sqrt[3]{\mathrm{BD}} + \tfrac{1}{4}'' \text{ for double belt} \qquad (38)$$

But Box[2] gives a simple rule, which seems to agree very well with the best practice, and his rule is

$$t \text{ in } \tfrac{1}{8}\text{ths} = \mathrm{D} \text{ in feet} + d \text{ in inches} + 5 \qquad (38\mathrm{A})$$

where D and d are diameters of pulley and shaft respectively. The length of the boss is, usually, for a *fast pulley* about $\frac{2}{3}$B, and for a *bushed loose pulley* equal to B, where B is the breadth of the pulley rim. For details of keys refer to Art. 93.

EXERCISES.

Design, etc.

1. A cast-iron pulley, 5′ in diameter, is to be designed for an 8″ single leather belt; the arms are to be straight, and segmental in section. How many arms would you arrange for? and what size would you make them at the boss and at the rim?

2. The skin stress in the shaft for the pulley in the previous question is 5000 lbs. per sq. inch, and half the power transmitted by the shaft is transmitted by the wheel. What size shaft would be required? and what thickness would you make the metal of the wheel's boss?

Sketching Exercises.

3. Sketch a belt gear suitable for driving a variable speed machine from a main shaft.
4. Make a sketch of a safety cap for a gib-headed key.
5. Make a sketch of a pair of fast and loose pulleys, suitable for use on the shaft of a machine. Be careful to show how the journal of the loose pulley is lubricated.
6. Sketch a belt gear for a slow forward and quick return motion, suitable for driving a screw-worked planing machine.
7. What is the object of curving pulley arms? When the arms are made straight, what precautions should be taken in designing and in casting?
8. Sketch a cast-iron split pulley, showing—
   (*a*) The joint when it occurs through the arms.
   (*b*) The joint when it occurs between arms.

Drawing Exercise.

9. Make working drawings of the pulley with curved arms (Figs. 566 and 567). Give t a suitable value for a single belt.

[1] "Machine Design," vol. i. p. 487.
[2] Box's "Mill Gearing," p. 98.

CHAPTER XXI

PISTONS AND CYLINDERS, ETC.

228. Function of a Piston.—A piston is a cylindrical body fitted to a hollow cylinder in such a way that although free to slide in it under the action of fluid pressure, as in a steam engine (Fig. 570), or if acting against fluid pressure under the action of a force, as in a pump (Fig. 571), practically no escape of the fluid from one side of the piston to the other takes place. Usually, when a **piston** is used as part of a pump, and it is provided with a valve or valves, which allow the fluid to pass from one side to the other of the piston during one of its strokes, it is called a **bucket**. Ordinarily the piston is attached[1] to a rod called the **piston rod**, which passes through a stuffing box in the cover of the cylinder in which the piston works, and is used to connect the piston to some piece outside the cylinder. If the piston be reduced in size or the piston rod increased, until they are the same size, we have what is technically called a **plunger**, as shown in Fig. 572, which forms an important part of the feed pump.

Small pistons (and large ones for the engines of cargo ships), where weight is of little importance, are made of cast iron. But in engines for large passenger ships, and all large warships, the pistons are made of *cast steel*, while in torpedo boats they are made of *forged steel*.

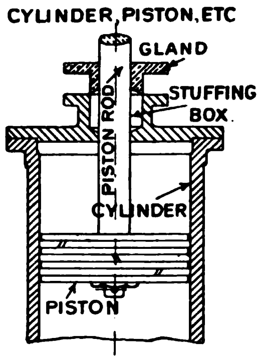


Fig. 570.

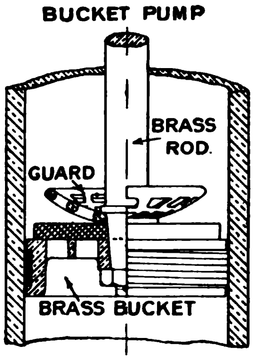


Fig. 571.

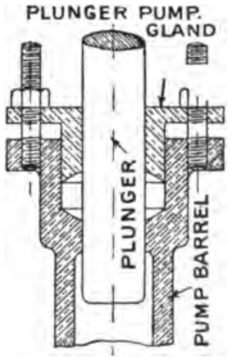


Fig. 572

229. Pistons without Packing.—For some purposes a plain cylindrical piston, Fig. 573, accurately fitting the cylinder, answers very well, particularly in cases where the resistance due to the packing would be objectionable, as with the piston used in the *steam engine indicator*, which is without packing, but is grooved, as in Fig. 574, to diminish leakage,[2] and to some extent lubricate the rubbing surfaces. This type of piston is sometimes used in pumps, and when the piston is sufficiently long there is very little wear.

[1] Occasionally, in small engines, the piston and rod are forged in one piece.

[2] The grooves present an abrupt change of section to any fluid passing by them, and impose a resistance to the flow which is, no doubt, in part due to the decrease of fluid pressure which occurs at each groove. There is very much to be said in favour of solid pistons, and there seems no satisfactory reason why, with more accurate work, suitable fits, large bearing surfaces, and highly finished surfaces, they should not be more generally used, particularly for vertical engines.

230. Piston Packings.—A plain or solid piston (one without packing), let it be ever so well proportioned or fitted, sooner or later becomes leaky, so, to prevent this, the pistons of heat engines are packed with metallic spring rings, many forms and arrangements of which are in use; indeed, of pistons of steam engines alone, there are endless varieties, due to efforts made in endeavouring to produce a perfect piston to stand the high speed and pressures in common use now. Notwithstanding these efforts it is safe to assume that the last word has not been said in piston design.

A **good** or **ideal piston** should be designed and constructed in such a way that it is sufficiently strong, keeps steam-tight for a considerable length of time, has comparatively few parts, its bolts and nuts being secured against working loose, and the piston as a whole to work silently with as little friction as possible; steam must not pass the rings at their joints, or find an easy passage beneath the rings and add its pressure to the normal spring of the rings against the cylinder walls, for if this occurred, the **3 or 4 lbs. per square inch between the rubbing surfaces**, which is found to be all that is required when the rings fit the cylinder accurately, would be increased to some ten to fifty times that amount, with consequent **undue wear** and **great loss of power**. Indeed, the pressure of the rings against the cylinder should never exceed what is allowable between rubbing surfaces under the conditions of lubrication and speed which prevail. A skilfully designed piston, worked with ordinary dry steam, should be efficiently lubricated by the condensation of the steam on the rubbing surfaces alone; but of course all pistons are **lubricated with high-flash oil when the steam is superheated.**

With these points before us it will be convenient to deal with the characteristic features of the best known pistons by grouping their like parts and features together. Commencing with the **packing spring rings**, the simplest of these, used for locomotives and other quick-running engines, are **Ramsbottom's**. They are made either of very tough close-grained cast iron, steel, gun-metal, or Perkins'[1] anti-friction metal; cast iron (which works better than steel on the cylinder surface) being most commonly used for cylinders of all sizes, and steel for locomotives. These rings are made rectangular in section, as shown in the five Figs. (for small and medium size pistons) 575 to 579, to fit separate grooves turned in the solid piston, there being two, three, or more grooves

PISTON PACKINGS, ETC.

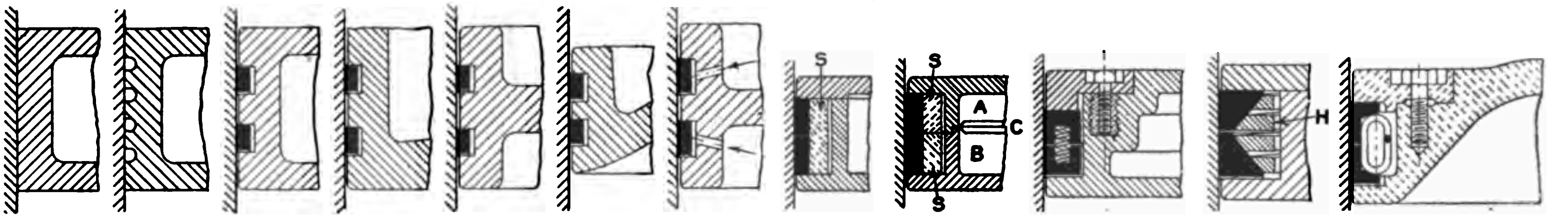


Fig. 573. Fig. 574. Fig. 575. Fig. 576. Fig. 577. Fig. 578. Fig. 579. Fig. 580. Fig. 581. Fig. 582.—Mudd's. Fig. 583. Fig. 584.—Buckley's.

according to the fluid pressure and depth of piston. In the simplest arrangements the rings are turned solid to a diameter about $\frac{1}{10}$ greater than that of the cylinder, and they are cut and the ends shaped to overlap, as in Fig. 602, or a piece is cut out and the

[1] Refer to Art. 177.

ends filed to an angle, as in Fig. 599. They are then carefully sprung over the piston into position,[1] and when the piston is placed in the cylinder they press against the bore or walls with sufficient force to make a steam-tight joint. Fig. 579 shows an arrangement for admitting steam to the back of the rings (to increase their fluid tightness), which has been occasionally used, but this principle, which answers so well in the leather collar of a hydraulic press, does not appear to be favoured by those who have tried it.

The **size** *of these* **rings** may be: **thickness** from 0·025D to 0·03D; and **width** 0·03D to 0·06D, where D is the diameter of the piston. Rings fitted in the way described are, when sprung into position, obviously not circular in form, and until they have become well bedded by wear will allow a small leakage to occur. For an ordinary steam cylinder this is of trifling importance, but in an internal combustion engine it would mean a loss of compression, and rings for such cylinders should be, after they are split and the ends drawn together, turned on their outer edge so that they exactly fit the cylinder.[2] Even then there is not an equal pressure between ring and cylinder wall all round, as this can only be secured by making the ring of varying thickness.[3] An approximation to this ideal condition is sometimes secured by making the **ring eccentric** in form, the thickest part being opposite the joint, and 1½ times thicker than the thinnest part. One of the disadvantages of Ramsbottom's rings is that they cannot be got at for removal without drawing the piston. To overcome this objection, and to allow of larger rings being used than could be sprung over the body of a piston, a **junk ring**[4] is used, as shown in Fig. 585, and this ring is sometimes made for large pistons of the form shown in Fig. 587 to overcome the above objection. The figure shows four eccentric rings at their thickest sections. In small engines sometimes the junk ring takes the form of a cover, which is held on to the piston by the piston rod nut, as shown in Fig. 580, or the piston itself is made in two parts A and B, Fig. 581, with the joint at the centre C, the parts being held together by the piston rod, as in the previous one.[5] It will be noticed that these two pistons have their rings backed up by **spring rings** S behind them, which are fitted in the same way, the pressure they exert on the outer rings makes the latter more effectively steam-tight; and Fig. 592 shows how the rings may be reinforced by the acting of a **spring** S and wedge W. For pistons of larger size these spring rings are not very efficient, and the old method (not often used in new work) of dealing with the problem was to press the piston ring against the cylinder wall by a number of dished springs or **coach springs** S, Fig. 590. The chief objection to this arrangement is that it is not possible to set all the springs so that the pressure on the ring is uniform, and being always in motion whilst the engine is running, the springs tend to wear themselves (as well as the piston) away, and furthermore, in horizontal engines it is important that the piston ring shall follow the sides of the cylinder freely, or, in other words, **float**, which it cannot do if the springs react from the piston body; moreover, the range of action of this form of spring is very limited. To

[1] In the case of cast-iron and anti-friction rings, this must be done with great care to prevent them breaking. No matter how carefully they have been jointed, there is often a slight leakage at the joint. To reduce this, the joints in the different rings should be kept as far apart as possible. Stop pins are sometimes used *near the ends* to keep them in such positions.

[2] Of course, this is always done with larger pistons fitted with junk rings. A piece of paper is placed between the junk ring and piston ring or rings, and the whole is screwed up tight, gripping the piston rings so that they can be turned and accurately fitted to the cylinder, after which of course the paper is withdrawn. Of course, with rings for the solid pistons referred to above, they are held in a suitable lathe chuck.

[3] Refer to Unwin's "Elements of Machine Design," Part II. p. 255.

[4] The early engineers, in dealing with *steam at low pressures*, packed their pistons with hempen rope soaked in tallow, which they called junk, and they used a ring to tighten it up as it became worn, or to allow of its being removed without drawing the piston. And although we now use for steam purposes metallic packing, we retain the name of the ring.

[5] In all these pistons the rings must be fitted, either by very accurate turning, by scraping, or by grinding, so that they are steam-tight between flange and ring, although free to move between them.

overcome these objections a number of expedients have been employed, thus in the **Cameron** piston, Fig. 591, we have a *corrugated ribbon of steel* pressing the piston ring out, the lateral or radial pressure being obtained by the resistance of the spring to being bent into a circle, and by the almost uniform pressure exerted by the corrugations when the ends of the springs are pressed apart, by packing pieces between them, without the spring touching the body of the piston. With this arrangement the piston ring can be comparatively thin, enabling it to conform to the shape of the cylinder when worn. Another arrangement, similar in principle, which is very generally used, particularly for low-pressure marine pistons, is **Buckley's**, shown in Fig. 584 (fitted to a steel piston).

FURTHER EXAMPLES OF PISTON PACKINGS.

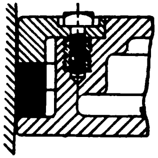


Fig. 585.

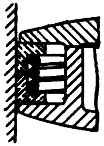


Fig. 586.

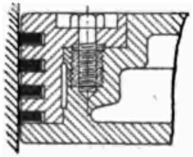


Fig. 587.

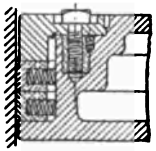


Fig. 588.

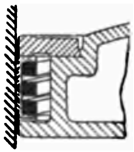


Fig. 589.

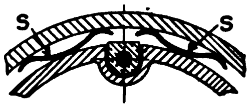


Fig. 590.—Coach springs.

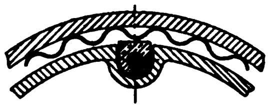


Fig. 591.—Cameron's.

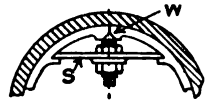


Fig. 592.

A flattened helical spring is so placed behind the inner slanting surfaces of two rings that it presses them out against the walls of the cylinder, and also down and up against the flange of the piston and of the junk ring. The same action is provided for in Fig. 586, where a helical spring of circular section is used. In fact, it has long been understood that *wear not only takes place between the ring and cylinder walls, but also between the edges of the ring and the flange of the piston and of the junk ring*, a very slight amount of play soon developing into a large degree of slackness, due to the continual concussion on change of motion at each stroke. Thus we have, in Fig. 583, **Clayton's and Goodfellow's** piston for mill engines, with a spiral spring H made of strong cast iron, and cut out of a ring of the metal, having four or five turns, being coiled inside the piston rings, and so shaped that, by wedging, it acts in the double way just described. **Mather & Platt's**, Fig. 589, is another piston where this principle in a modified form is employed, the spiral hoop or spring being sometimes made of steel. These forms have on the whole given much satisfaction, the objection sometimes raised against them, that no adjustment of the spring is possible, and that therefore it is always exerting its maximum effect, does not appear to be an important one. It should be understood that the chief part of the elasticity of the spring is exerted in pressing the rings against the junk ring and flange, and that the friction so caused helps to prevent undue pressure on the cylinder walls when first fitted, and there is sufficient of it when the cylinder is worn. Closely allied to these in principle is **Mudd's** arrangement, shown in Fig. 582, two rings each 2″ × 2″ (for all sizes of pistons from 18″ to 80″ diameter) are pressed against the flange and junk rings by a number of helical springs, placed as shown, and separate helical springs (not shown) are applied tangentially at the joints of the rings to press them against the cylinder walls.

230a. Allen's Piston.[1]—This invention is a radical **departure** from ordinary practice **of great promise**. The **rings** are each **made in three pieces**, and their opposing ends are carefully fitted to cast-iron expanding pieces, the inner ends of which fit into holes drilled radially into the piston. Internal springs keep these pieces up to their work, and thus furnish the pressure to hold the rings against the cylinder walls. Should experience prove that pistons of this type are free from internal trouble when worked by

[1] Manufactured by Messrs. Allen and Simmons of Reading.

superheated steam or gases of high temperature, there will probably be a great future for them and pistons of this type, notwithstanding their cost.

231. Pump Bucket Packings.—The buckets of air pumps (and occasionally circulating pumps[1]) are packed either with wood staves or with cotton or hemp, usually with the latter. When wood is used, lignum vitæ is preferred, as it works well with the brass barrel. Figs. 593 and 594 show how small buckets are sometimes fitted; in the latter we have two rings, each being made up of a number of blocks, breaking joint with those in the other ring. Fig. 595 shows, for larger buckets, another way of breaking joint to prevent leakage; the staves A have keys B let in them where the joints occur, and two stiff rubber tubes D with a distance ring C are squeezed in to act as an elastic cushion at the back to keep the bucket tight as wear takes place. When wood packing is used it must be very carefully fitted, and there must be clearance enough to allow the wood to swell when it is wet. A simpler and very effective packing is the hemp or cotton rope one, Figs. 596 and 597. It will be seen that the flanges of the bucket have been grooved and drilled at PP to allow of the ends of the rope being pegged in and held secure. In Fig. 598 a junk ring is used, but it is not intended to adjust the packing, but merely to keep it in position; it is screwed down firmly to the bucket. The depth d of the packing is about 0·1 diameter D + 1·4, and the thickness t of the packing $\frac{d}{4}$ to $\frac{d}{5}$.

PUMP BUCKET PACKINGS.

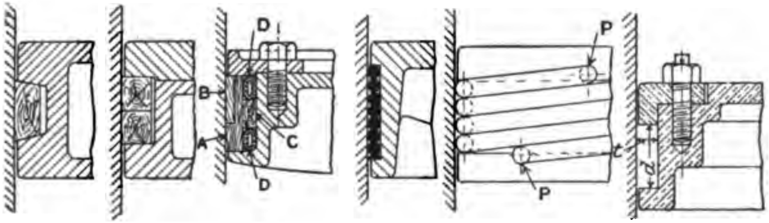


FIG. 593. FIG. 594. FIG. 595. FIG. 596. FIG. 597. FIG. 598.

232. Piston Ring Joints.—We have seen that in small rings the joint is made as shown in Fig. 599, and that, when the rings are made to break joint, very little steam passes them. But with large rings, some kind of tongue or stop piece, offering a barrier

PISTON RING JOINTS.

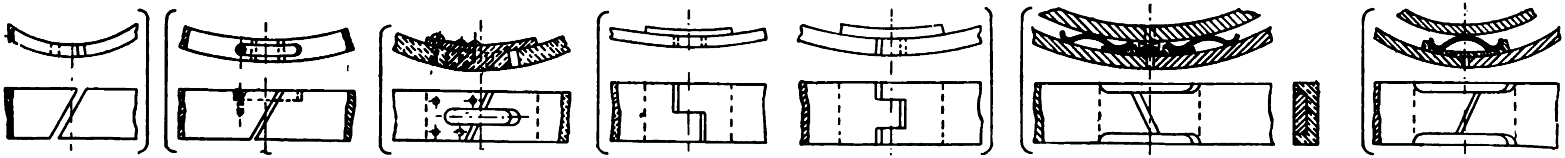


FIG. 599. FIG. 600. FIG. 601. FIG. 602. FIG. 603. FIG. 604. FIG. 605.

at the joint to the passage of steam, is used; in Figs. 600 to 605 several of these are shown; they are made of brass and are screwed to the ring by countersunk headed screws, and the drawings should speak for themselves.

233. Guard Rings and Devices.—Every nut and screw about a piston must be so secured that it is impossible for it to work off

[1] There is not much necessity to pack these for Marine purposes, as the water usually flows freely into the pump by gravity, and the pump runs too fast to allow of much leakage past the piston if the latter be well filled. Thus the frictional resistances are sensibly reduced when no packing is used.

and do serious damage to the cylinder cover, or to the piston itself. Some of the fittings used to secure or **lock nuts**, etc., are shown in Figs. 606 to 612. One of the simplest is to recess the junk ring, as in Fig. 606, so that the nuts are flush with the top of the ring, and then to prevent rotation of the nut by a brass screw in contact with one of its sides, and screwed firmly home into the junk ring.

Figs. 607 to 611 show different forms of **guard rings**, most of which are held to the junk ring by small studs fitted with split

GUARD RINGS AND DEVICES.

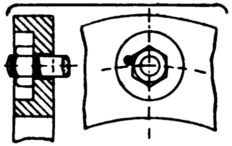


FIG. 606.

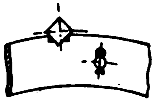


FIG. 607.

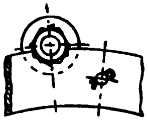


FIG. 608.

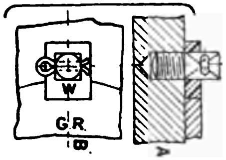


FIG. 609.

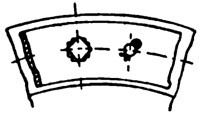


FIG. 610.

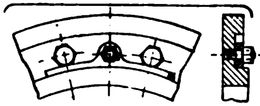


FIG. 611.

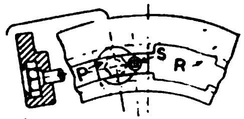


FIG. 612.

pins. Sometimes these studs are made (as they should be) with square shoulders, fitting in square holes in the rings to prevent rotation. Fig. 609 shows how the stud itself is in some cases made with the upper part square, the guard ring GR preventing rotation of the washer W, into which the stud fits.

A neat, but rather expensive, arrangement is shown in Fig. 612, where a dove-tailed plate P is placed in the recess R and slid into position over a nut towards the left, and held there by the brass screw S, the circle around the holding-down screw showing the clearance for a box spanner. The other devices shown should need no further explanation.

234. Connection of Piston to Piston Rod.—Small pistons that are not likely to require removing from their rods are sometimes

FIXINGS OF PISTONS TO RODS.

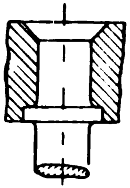


FIG. 613.

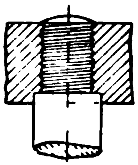


FIG. 614.

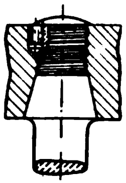


FIG. 615.

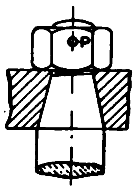


FIG. 616.

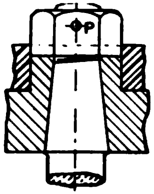


FIG. 617.

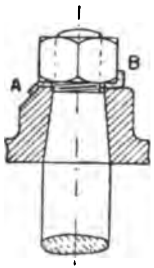


FIG. 618.

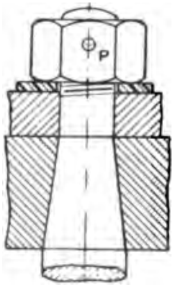


FIG. 619.

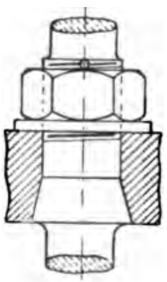


FIG. 620.

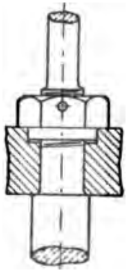


FIG. 621.

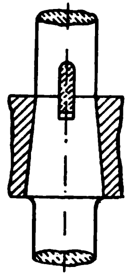


FIG. 622.

attached to the latter, as shown in Fig. 613, the rod being accurately turned to fit the hole in the piston and riveted over, forming a

countersunk head. Another simple connection is the screwed one, Fig. 614. The end in this case is also slightly riveted, but only enough to prevent the rod unscrewing. A combined cone and screwed end is shown in Fig. 615, with the set-screw shown to prevent unscrewing. Figs. 616 to 619 show forms often used, the one in Fig. 618 having its nut held by a safety washer, one wing A of which is in contact with a flat on the side of the piston boss, and the other B with a side of the nut. The nuts in Figs. 616, 617, and 619 are secured by split taper pins, and these are also used in Figs. 620 and 621, which show how pistons are fitted to *tandem* rods. Fig. 622 also shows a tandem rod, but the piston is held on in this case by a cotter. It will be noticed that the angle of the taper or conical parts varies considerably; in practice the taper ranges from about 1 in 4 to as little as 1 in 20. Piston rods are occasionally secured to large pistons, as shown in Fig. 623, by a flange, which enables the piston to be very easily drawn, but the nuts require well locking.

235. Proportions of Cast-iron Pistons.—A very simple form of cast-iron piston which is largely used for diameters up to about 16″ is shown in Fig. 624, and the following Table 10 gives suitable dimensions. It will be seen that the junk ring takes the form of a cover, which

PROPORTIONS OF CAST-IRON PISTONS.

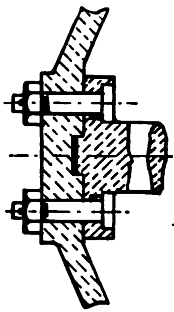


FIG. 623.

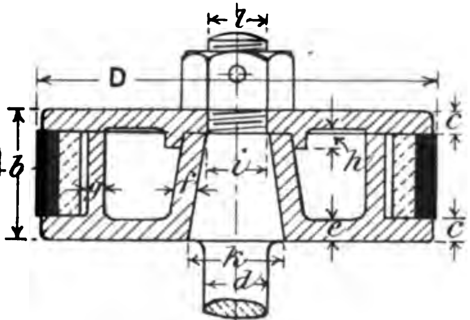


FIG. 624.—Small cast-iron piston.

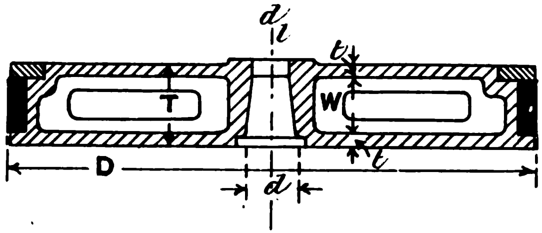


FIG. 625.—Large cast-iron piston.

TABLE 10.—DIMENSIONS OF CAST-IRON PISTONS[1] UP TO 16″ DIAMETER. (Fig. 624.)

Diameter of cylinder D.	*b*	*c*	*d*	*e*	*f*	*g*	*h*	*i*	*k*	*l*
6	3	$\frac{1}{2}$	$1\frac{1}{8}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{5}{16}$	$\frac{5}{16}$	$1\frac{1}{4}$	$1\frac{3}{4}$	1
8	$3\frac{1}{4}$	$\frac{9}{16}$	$1\frac{3}{8}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{8}$	$\frac{5}{16}$	$1\frac{3}{8}$	2	$1\frac{1}{4}$
10	$3\frac{5}{8}$	$\frac{7}{8}$	$1\frac{5}{8}$	$\frac{9}{16}$	$\frac{11}{16}$	$\frac{3}{8}$	$\frac{3}{8}$	$1\frac{5}{8}$	$2\frac{3}{8}$	$1\frac{1}{2}$
12	4	$\frac{11}{16}$	2	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{1}{2}$	$\frac{3}{8}$	$1\frac{7}{8}$	$2\frac{3}{4}$	$1\frac{7}{8}$
14	$4\frac{1}{2}$	3	$2\frac{1}{4}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{1}{2}$	$\frac{3}{8}$	$2\frac{1}{8}$	$3\frac{1}{4}$	2
16	$4\frac{3}{4}$	$\frac{3}{4}$	$2\frac{3}{8}$	$\frac{11}{16}$	$\frac{7}{8}$	$\frac{1}{2}$	$\frac{3}{8}$	$2\frac{5}{16}$	$3\frac{5}{8}$	$2\frac{1}{4}$

[1] Haeder and Powell's "Handbook on the Steam Engine."

is held in position by the piston rod nut, as explained in connection with Fig. 580. Fig. 625 shows a section of a **large cast-iron piston.** This type is often fitted with Ramsbottom rings, as shown in Fig. 587. The thickness t of the ribs and the upper and lower walls may be from about $\frac{D}{60} + 0{\cdot}4''$ to $\frac{D}{40} + 0{\cdot}4''$, the lower value being for the low-pressure cylinder. To enable the walls to be strong enough between the stiffening webs or ribs W, the number N of the latter should be at least as follows :—

$$N = 4 \text{ for } D = 12'' \text{ to } 24'' \qquad N = 6 \text{ for } D = 24'' \text{ to } 40''.$$
$$N = 8 \text{ for } D = 40'' \text{ to } 60'' \qquad N = 10 \text{ to } 12 \text{ for } D = 60'' \text{ to } 80''.$$

And the other main proportions may be as follows :—
$d_1 = 1{\cdot}5K$ to $1{\cdot}7K$, $T = 1{\cdot}4K$ to $1{\cdot}7K$ (for values of K refer to Table 11). Diameter of junk ring bolts $= 0{\cdot}1C + \frac{1}{4}''$; pitch of junk ring bolts, about 10 diameters.

The coefficient
$$C = \sqrt{p} + 1 \times \frac{D}{50}$$

Where $p =$ *half boiler pressure*[1] for high-pressure pistons, *quarter boiler pressure* for medium-pressure pistons, and boiler pressure $\div$ ratio of low pressure to high pressure piston diameters for low-pressure pistons.

The thickness of **locomotive pistons** usually = diameter × 0·28.

TABLE 11.—VALUES OF COEFFICIENT K FOR PISTONS.[2]

(Admission pressures in lbs. per sq. inch, absolute.)

Diameter of cylinder D.	Pressure 15 to 30.	Pressure 30 to 55.	Pressure 55 to 85.	Pressure 85 to 114.	Pressure 114 to 142.	Pressure 142 to 170.	Pressure 170 to 200.	Pressure 200 to 227.
Inches.	Inches.	Inches.	Inches.	Inches.	Inches.	Inches.	Inches.	Inches.
15 to 23	1	$1\frac{1}{8}$	$1\frac{3}{8}$	$1\frac{1}{2}$	$1\frac{3}{4}$	2	$2\frac{1}{8}$	$2\frac{1}{2}$
23 to 31	$1\frac{3}{8}$	$1\frac{1}{2}$	$1\frac{3}{4}$	2	$2\frac{1}{2}$	$2\frac{3}{4}$	3	$3\frac{3}{8}$
31 to 39	$1\frac{1}{2}$	$1\frac{3}{4}$	2	$2\frac{1}{2}$	$3\frac{1}{8}$	$3\frac{1}{2}$	$3\frac{3}{4}$	4
39 to 47	$1\frac{3}{4}$	2	$2\frac{3}{8}$	3	$3\frac{1}{2}$	4	$4\frac{1}{2}$	
47 to 55	2	$2\frac{1}{4}$	$2\frac{3}{4}$	$3\frac{1}{8}$	$3\frac{3}{4}$	$4\frac{1}{8}$		
55 to 63	$2\frac{3}{8}$	$2\frac{1}{2}$	$3\frac{1}{8}$	$3\frac{1}{2}$	4	$4\frac{3}{8}$		
63 to 71	$2\frac{1}{2}$	3	$3\frac{3}{8}$	4	$4\frac{3}{8}$			
71 to 79	$2\frac{3}{4}$	$3\frac{1}{8}$	$3\frac{1}{2}$	$4\frac{1}{8}$				
79 to 86	3	$3\frac{3}{8}$	$3\frac{3}{4}$	$4\frac{3}{8}$				
86 to 94	3	$3\frac{1}{2}$	4					
94 to 102	$3\frac{1}{8}$	$3\frac{1}{2}$						
102 to 110	$3\frac{3}{8}$	$3\frac{3}{4}$						

[1] Seaton's "Marine Engineering."

[2] Dr Bauer's "Marine Engines and Boilers."

236. Cast-steel Pistons.—We have explained (Art. 228), that in cases where weights must be kept down, cast steel pistons are used. They are made conical in form, as shown in Figs. 626 and 627, which gives strength and rigidity, so this, with the additional advantage of the stronger material, reduces their weight some 30 to 35 per cent. Fig. 626 shows such a piston arranged for *Ramsbottom's Rings* (shown more in detail in Fig. 587), the projecting shoulder P on the boss is useful in lifting the piston. To prevent rotation of the rod when screwing up, a snug S is used. Fig. 627 is a form which, when fitted with Buckley's packing (Fig. 584), is largely used for low-pressure pistons.

PROPORTIONS OF CAST STEEL PISTONS.

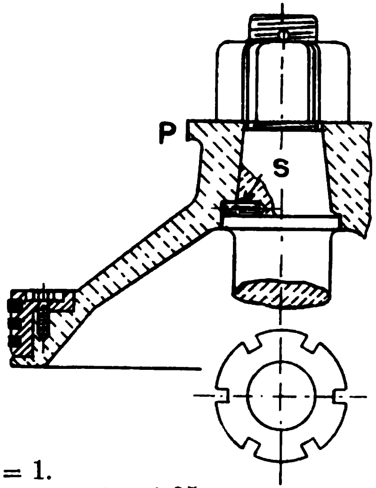


Fig. 626.

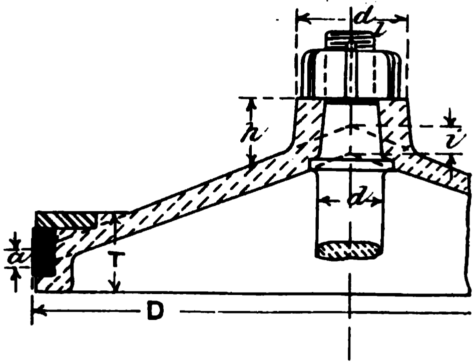


Fig. 627.

The following are **suitable proportions** for **cast-steel pistons** (for values of coefficient K refer to Table 11):—
Height of boss h (Fig. 627) = 1·1K;

diameter of boss d_2 = 1·7K for small pistons, and 1·5K for large ones and light engines.

The dimensions h, b, and t, in engines with several cylinders are usually made the same for all pistons.

The thickness i, measured on the centre line, $= K \times c$, the following being the values of the coefficient c:—

For flat pistons, or for inclinations from 0° to 6°, $c = 1$.
For slightly covered pistons inclination from 6° to 18°, c = 0 85 to 0·95
" " " " " 18° to 28°, c = 0·75 to 0·85
" " " " " 28° to 35°, c = 0·65 to 0·75

And the thickness a measured at the side of the piston = 0·45i to 0·55i.

237. Cylinder for Steam Engine (Drawing Exercise).—In Fig. 628 is shown the elevation and sectional plan of a cylinder for a horizontal steam engine; with detail drawings of the cylinder covers, slide valve, glands, and bushes. As a **simple exercise**:—

Instructions.—*Draw* the section of cylinder on AB, assembling the slide-valve covers, glands, etc., as in actual use. **Scale 3 inches = 1 foot.**

238. The following is a **more advanced exercise**:—

Instructions.—*Draw* the section of the cylinder on AB, with *all* parts assembled in position: *and* in projection from it a vertical section, taken on the line EF. Fitting the valve case with a suitable cover. **Scale 3 inches = 1 foot.**

239. Pistons for Internal Combustion Engines.—These pistons are made of cast iron, and, as they are used with single-acting engines,[1] it is convenient to make them long enough to also act as a guide for the connecting-rod end, and to receive its angular

[1] Pressure on one side of the piston only.

DRAWING EXERCISE.

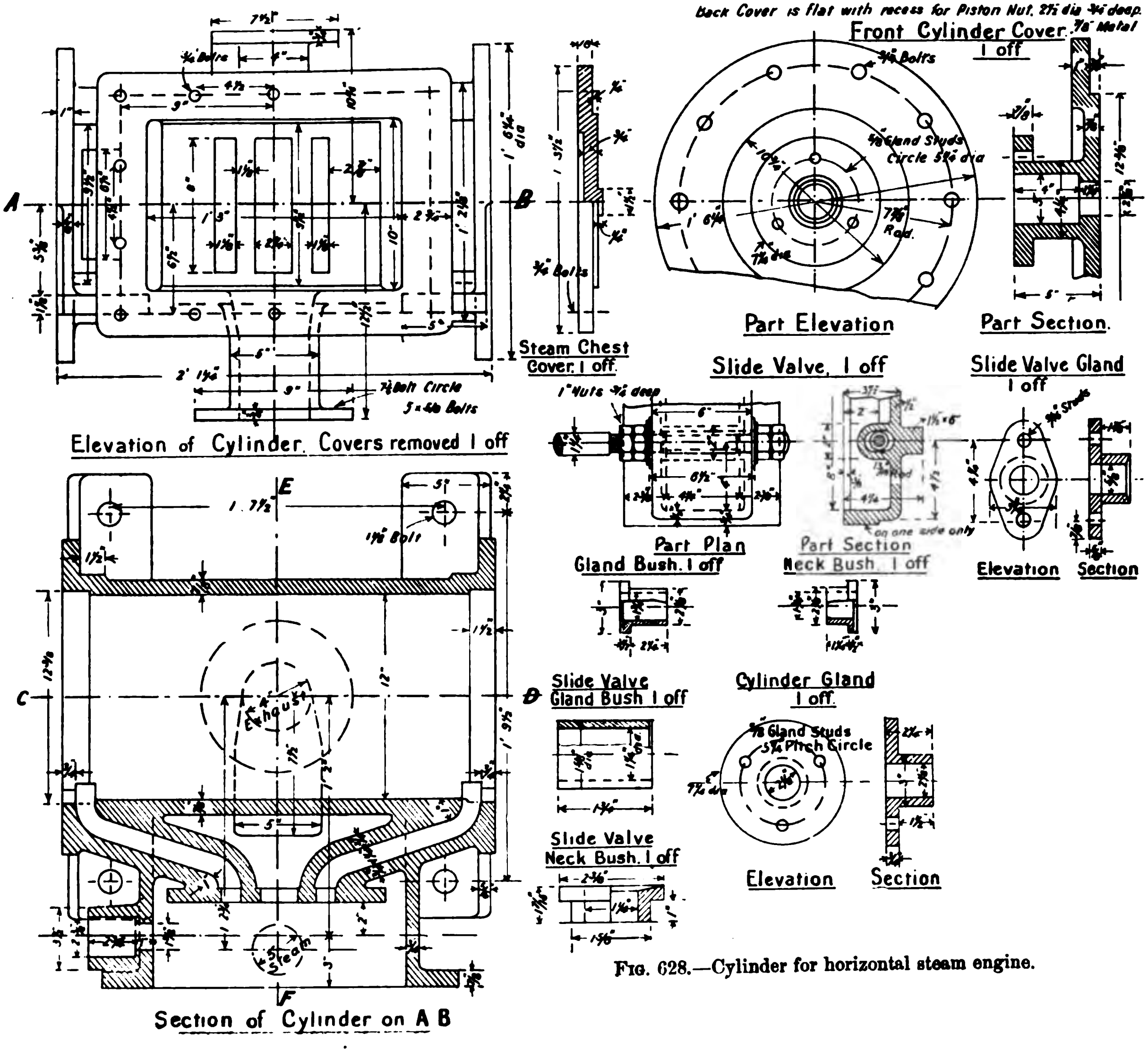


FIG. 628.—Cylinder for horizontal steam engine.

thrust. The pistons being hollow with open ends, they are commonly called *trunk pistons*. As the closed end during the explosion stroke is in contact with the burning gases it should be made a shade smaller[1] in diameter than the body, so that when at work it fits the cylinder uniformly from end to end. Fig. 629 shows a piston suitable for a *gas engine*. The usual average proportions being given in terms of D the diameter. For high-speed engines $L = D$ to $1{\cdot}6D$; for large engines $L = 1{\cdot}2D$ to about $1{\cdot}75$; whilst for small engines L ranges from $1{\cdot}4D$ to $2{\cdot}25D$. It is packed with *Ramsbottom's Rings*, whose number is usually about $\frac{D}{2}$, with a minimum number of three. Fig. 629A gives suitable average proportions for **pistons of petrol engines**. The **gudgeon pin** GP should

PISTONS FOR INTERNAL COMBUSTION ENGINES.

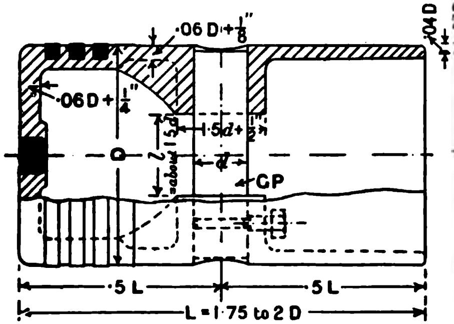


Fig. 629.—Gas engine piston.

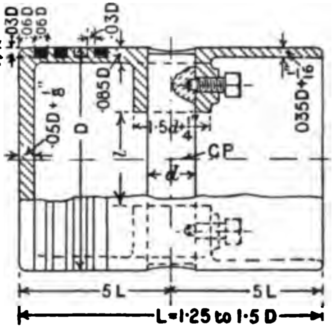


Fig. 629A.—Petrol engine piston.

have for both cases such diameters and lengths that the maximum pressure does not exceed 800 to 1000 lbs. per sq. inch of projected bearing.[2] And they should be checked for bending,[3] when the maximum stress for steel should not exceed 15,000 lbs. per sq. inch.

240. Piston for Petrol Engine (Drawing Exercise).—A plan and sectional elevation of a petrol engine piston are shown in Fig. 630.

Instructions.—*Draw* these two views, and a complete elevation. Also show in detail, by separate drawings, the set-screws and gudgeon pin. Scale full size.

241. Cylinders for Petrol Engine (Drawing Exercise).—An elevation and sectional elevation of a pair of cylinders for a petrol engine are shown in Figs. 631 and 632.

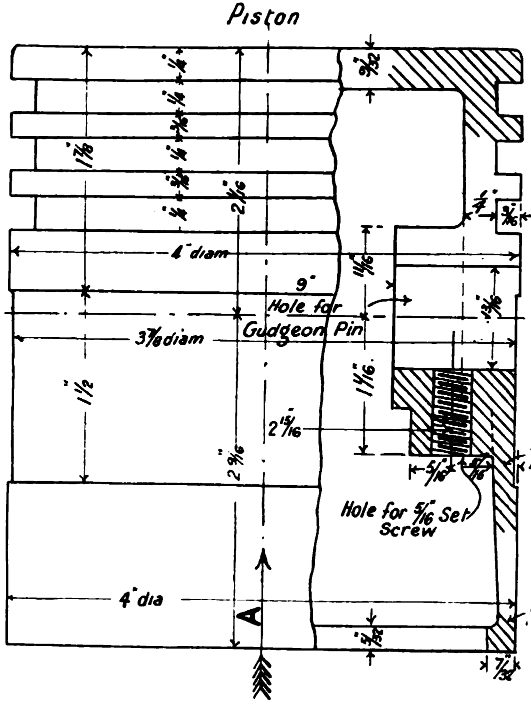


Fig. 630.—Piston for petrol engine (Drawing Exercise).

[1] About 0·01D" smaller.
[2] The area of the gudgeon pin is often twice that of the crank, the latter being subjected to a maximum pressure of 4000 lbs. per sq. inch.
[3] See author's "Machine Design, etc.," p. 510.

PAIR OF CYLINDERS FOR 20 H.P. FOUR-CYLINDER PETROL MOTOR (Drawing Exercise).

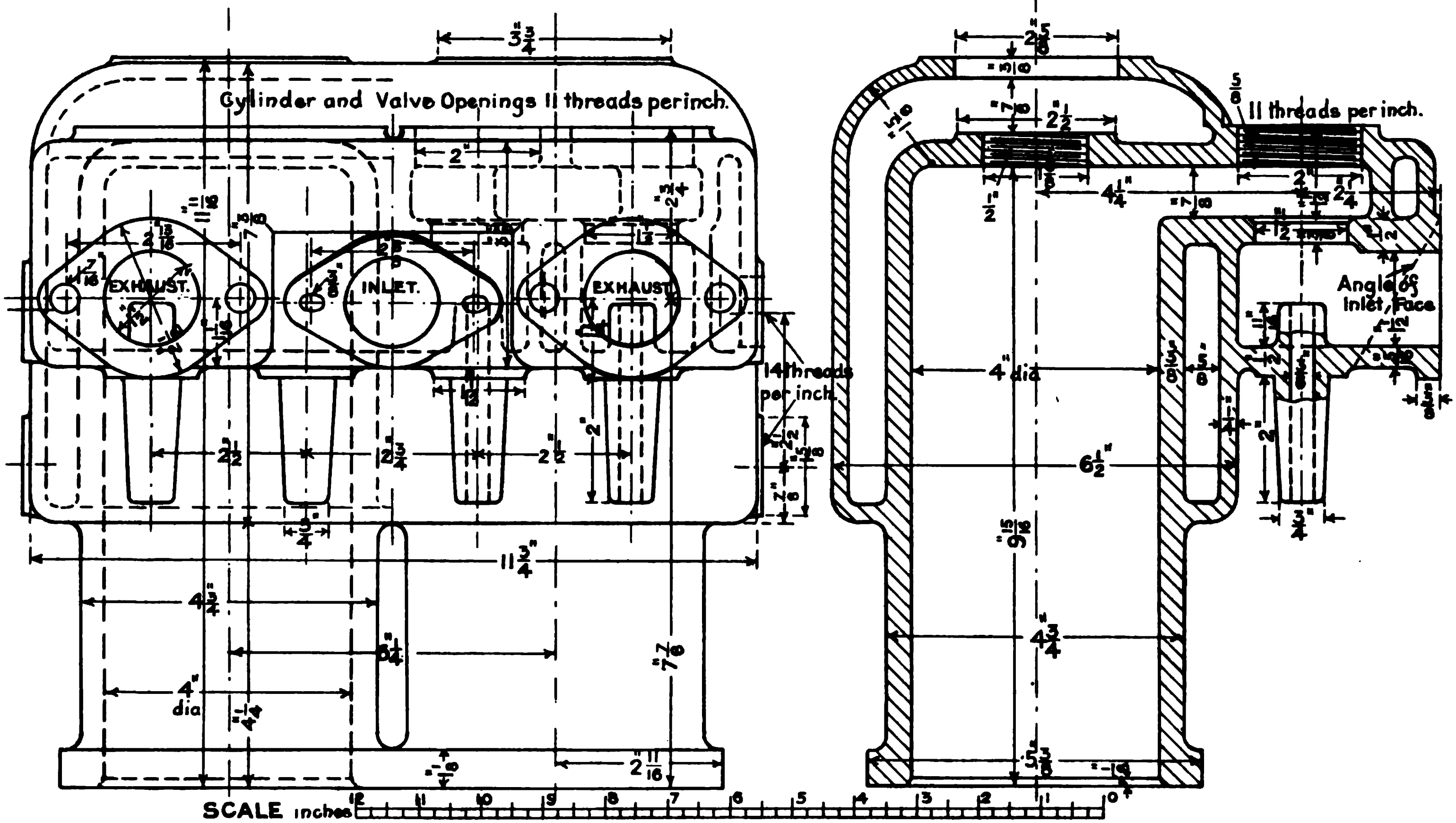


Fig. 631. Fig. 632.

Instructions.—*Draw* these views, and from the elevation project a sectional plan; half of the plan being a section through the exhaust and inlet openings, due to a horizontal cutting plane. **Scale full size.**

242. Piston Rods.—We have seen that usually piston rods at the piston end are formed with a taper and attached to the piston by a nut or cotter. In some types of engines, such as marine, we have the piston rod forged to the cross head, but when not made in this way the rod usually has a taper end fitted into the cross head, the large diameter of which is rather less than that of the body of the rod, which allows for the re-turning of the rod without interfering with the fit in the cross head. The material [1] in common use for piston rods is almost exclusively *Siemens-Martin Steel*, excepting for warships, when *crucible or nickel* steel is often used.

EXERCISES.

Design, etc.

1. Referring to the piston in Fig. 624, determine for one of 16″ diameter (by using the dimensions in the table of proportions) what the stress in the rod, and in the screwed portion of the rod, would be due to a pressure on the piston of 140 lbs. per sq. inch.
2. Make a sketch design of a petrol engine piston (Fig. 629A). Diameter 4¼″; the maximum pressure on the piston may be taken at 200 lbs. per sq. inch, and the maximum pressure on the gudgeon, 1000 lbs. per sq. inch. Choose your own working stress for the gudgeon pin.

Drawing.

3. Make working drawings of a cast-iron piston for a 16″ cylinder (Fig. 624). Scale half size.
4. Make complete drawings of a petrol engine piston. Diameter 5″, length of piston 7″, diameter of gudgeon pin 1¼″. Scale full size.

Sketching.

5. Show by sketches Mudd's, Buckley's, and Cameron's piston packing.
6. Sketch two different ways of packing an air-pump bucket to make it water-tight. What precautions must be taken when wood packing is used?
7. Show three different ways of arranging the joint in a piston ring to prevent leakage of steam past it.
8. Sketch three different ways of preventing the piston junk-ring screws from working loose.
9. You are to show by sketches how a piston is attached to its rod in the following cases:—
   (*a*) By *coning* the end of the rod and using a nut.
   (*b*) By flanging the end of the rod and bolting the piston on to it.
   (*c*) In the case of a *tandem* engine, attaching the piston to a coned part of the rod by means of a cotter.
10. Make a sketch of a small cast-iron piston.
11. Sketch a cast-steel conical piston, suitable for very large engines. What is the object of making a piston conical form? What angle of the cone gives the lightest piston?
12. Make a sketch of a piston suitable for a petrol engine.

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[1] Formerly piston rods were made of *scrap iron forging* of the highest quality, but this had a smaller strength, and was, owing to its fibrous character, liable to wear in *ridges or flutes*.

CHAPTER XXII

CROSS HEADS AND GUIDES

243. Cross Heads.—The part of an engine which connects together the piston rod and connecting rod is known as the *cross head.* It is so formed that it is guided or constrained to move in a straight line by the parts called **slides.** Cross heads are made in a great variety of forms in either wrought iron, mild steel, cast iron, or cast steel. Some representative examples of types used in stationary engines, locomotives, and marine engines are shown in Figs. 634 to 672. But, before we proceed to touch on these, attention may be given to the forces acting at the cross head.

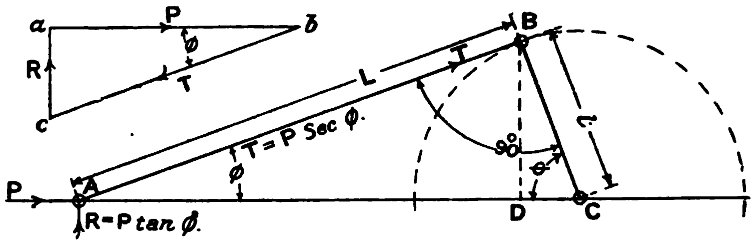


FIG. 633.

244. Forces acting at the Cross Head.—Fig. 633 is a diagrammatic drawing of a crank and connecting-rod arrangement. A being the cross head, AB the connecting rod, BC the crank, and R the reaction of the guide on the cross head, which is a maximum when the angle ABC = 90°; if the steam cut-off does not occur before that position is reached. Let P = the total pressure on the piston $= pD^2\frac{\pi}{4}$, p being the steam pressure per sq. inch, which we may assume to be constant in this case, and we may neglect the inertia forces. Then, the triangle of forces shown (in which ab represents P to a suitable scale) gives us the magnitude of R in terms of P, for $\frac{R}{P}=\frac{ac}{ab}$.

But, by similar triangles,

$$\frac{R}{P}=\frac{BD}{AD}=\frac{BC}{AB}=\frac{l}{L}$$

therefore—

$$\frac{R}{P}=\frac{l}{L}, \qquad \text{or } R=\frac{L}{Pl}. \qquad \text{Let } \frac{l}{L}=n, \qquad \text{then } R=Pn \quad \ldots\ldots\ldots\ldots \quad (39)$$

So, by decreasing the length of the connecting rod we increase the pressure R on the guides,[1] and this explains one of the

[1] For other relative positions of crank and connecting rod it can be shown that $R = P\frac{\sin\theta}{\sqrt{(n^2 - \sin^2\theta)}}$

objections to short connecting rods. And the above shows that for any angle ϕ the connecting rod makes with the horizontal $R = P \tan \phi$.

Furthermore
$$\frac{T}{P} = \frac{bc}{ab} = \frac{AB}{AD} = \text{secant } \phi \qquad \therefore T = p \text{ secant } \phi \qquad (40)$$

245. Position of Gudgeon or Cross Head Pin in Relation to Sliding Surface.—We have seen in the previous Article that in every case there is a certain position of the cross head which corresponds to its greatest pressure R on the guides. Now, obviously the best position for the axis C of the gudgeon pin, Fig. 634, in relation to the sliding surface DE of the cross head, is such that it is midway between D and E; the pressure is then evenly distributed over the sliding surface; but cases sometimes occur where it is convenient (but not good practice) to fix the gudgeon pin out of the centre. If this is done and C is over B, then it can be shown that the maximum pressure on the slide occurs at E, and that it uniformly tapers off to nothing at D. Or, of course, should C be over A, then the maximum pressure is at D, and there is no pressure at E. Fig. 635 shows a very bad case, the gudgeon pin overhanging the sliding surface, and the force R, acting through C, is tending to tilt the cross head about E, causing an upward pressure at D, and a bending action on the piston rod.

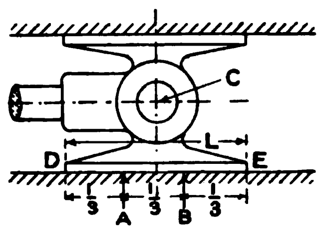


Fig. 634.—Correct arrangement.

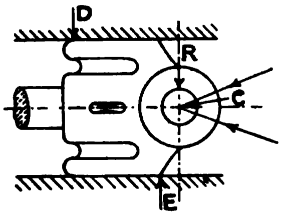


Fig. 635.—Showing defective arrangement.

246. Types of Cross Heads.—We have remarked upon the fact that there seems no end to the number of different forms that designers have given and are giving to cross heads, and it is hardly possible to classify them in such a way as to say that this or that particular design is a locomotive one or a marine one, as the case may be; but still, for our purpose it will be convenient to group them under the headings of stationary engine, locomotive, and marine. Commencing with the **stationary types**, we have, in Fig. 636, a simple and inexpensive form for small engines, the sliding surfaces being turned and bored. Although mostly used on cheap engines, there is an increasing tendency to use it in a more complete form, such as shown in Figs. 640 to 643 for important ones. In Figs. 640 and 641 the wrought-iron or steel rod is cottered into a cast-iron head, in which the brasses are held by wrought-iron or mild steel bolts and cap, the connecting rod having a forked end, in which is fixed the gudgeon. The head in Figs. 642 and 643 is fitted with cast-iron shoes S which are adjusted for wear by the cotters C, the gudgeon pin P being fitted with a Stauffer lubricator. Figs. 644 and 645 show an example of a cross head for two-bar guides, containing within itself a means of adjustment, namely, the nuts N and screws S, while the slide bars are fixed and properly arranged to resist the pressure. The head, which is used with a forked connecting rod, is made of malleable cast iron or cast steel, and the side blocks of cast iron. In Figs. 646 and 647 we have a **slipper cross head**,[1] both the piston rod and the cast-iron slipper being cottered to the wrought-iron head, which is bushed with gun-metal and fitted with a lubricator. A very simple and compact cross head of this type, suitable for small engines, is fully shown in Figs. 637 to 639. In a slightly different form it was an example in the Science and Art Examination Paper of 1893. A different

[1] Generally used only on stationary engines when they run in one direction only, so that the pressure is always on the bottom bar, and the slipper block can be arranged to run in a bath of oil.

CROSS HEADS, STATIONARY ENGINE TYPES.

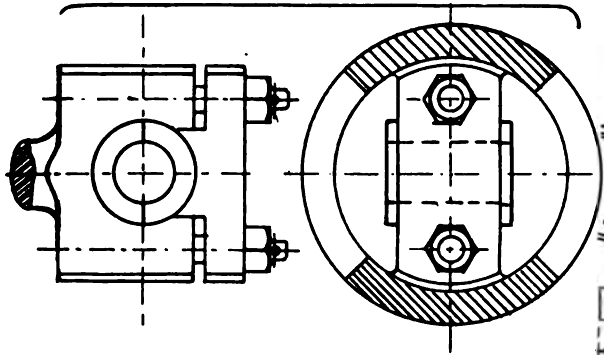


FIG. 636.—Piston rod head in cast-iron guide.

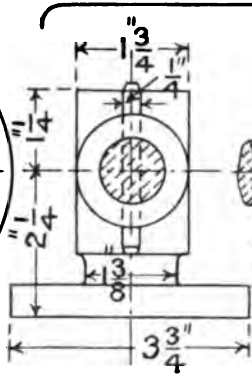
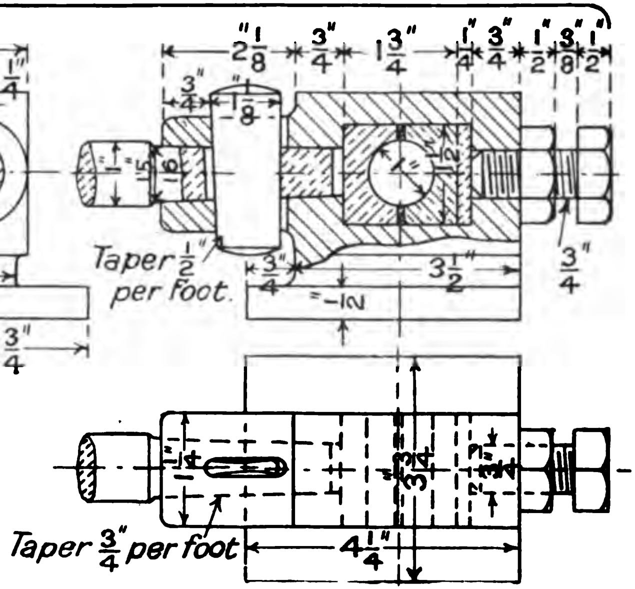
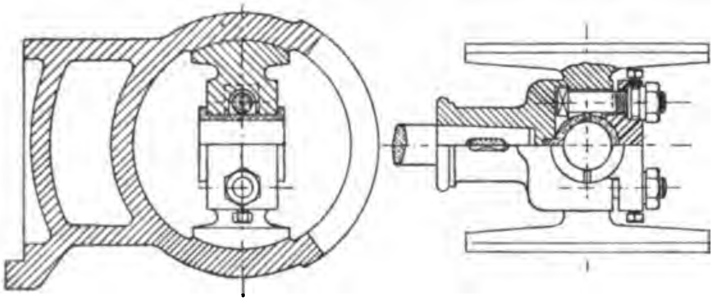


FIG. 637.

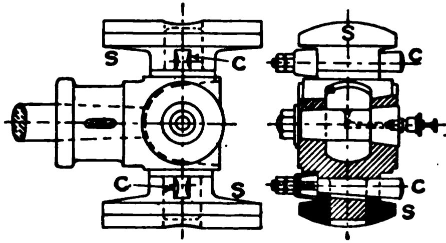


FIGS. 638, 639.—Slipper cross head for small engines (Drawing Exercise).

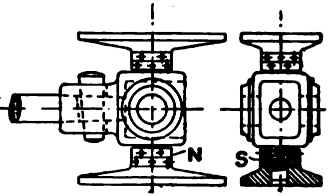
FURTHER TYPES OF STATIONARY ENGINE CROSS HEADS.



FIGS. 640, 641.—Cast-iron cross head.

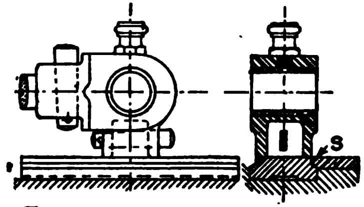


FIGS. 642, 643.—C.I. adjustable cross head.



FIGS. 644, 645.—Adjustable cross head.

type is shown in Figs. 649 to 653, four slide bars S being required; the forked head is of wrought iron or steel cottered to the piston rod, and the gudgeon pin P, passing through it, forms a neck journal for the connecting rod, and also two end journals J, on which fit the cast-iron slide blocks B. The guide bars are notched at the end E, so that the slide block passes the edge of the notch each stroke for even wear.[1] Figs. 654 to 656 also show a slipper cross head, the cast-iron slipper S being screwed to the wrought-iron head, whilst the gun-metal steps are tightened up by a side cotter, adjusted by the screw A, as shown. A more important and expensive slipper cross head (of the **marine type**) is shown in Figs. 657 to 659; the piston rod and head are in one forging, and the cast-iron slipper block is screwed to a plate A, which is dovetailed into the head. A four-bar **locomotive** cross head is shown in Figs. 660 and 661, and a two-bar one in Figs. 662 and 663, whilst Figs. 664 and 665 show an original and interesting cross head designed by Mr. W. Adams of the G.E.R. The cast-iron head is made in two parts bolted together by eight $\frac{7}{8}''$ bolts. It will be seen that the steel slide bar is drilled to allow the oil to reach the under side. The maximum pressure allowed on the sliding block is about 40 lbs. per sq. inch. Figs. 666 to 668 show still another locomotive cross head, but of the slipper[2] kind, designed by Mr. Stroudley. The wrought-iron head is forged in one with the piston rod. The pin is also wrought-iron but case-hardened, the steps being of gun-metal.[3] The ordinary direct-acting **marine** *engine* is usually so arranged that the gudgeon pin for the cross head is secured to the forked end of the connecting rod, and it works in a bearing in the cross head, as we have seen in some previous examples, and as shown in Figs. 669 and 670. It will be seen that the steps and slipper block[4] of the cross head in Figs. 671 and 672 (which is largely used), **are faced with white metal** and the other details of construction should speak for themselves.



FIGS. 646, 647.—Slipper cross head.

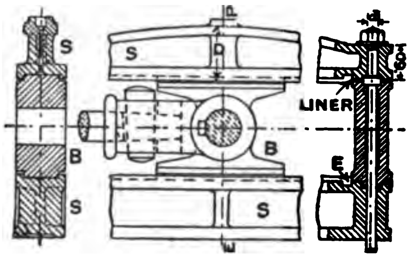


FIG. 649.—Section on line ER. FIG. 650. FIG. 651.—Section on line GH.

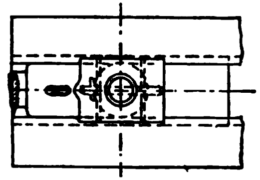


FIG. 648.—Plan.

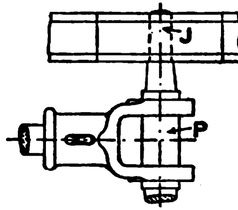


FIG. 652.—Plan. Four-bar stationary cross head.

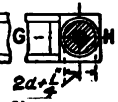


FIG. 653.

247. Cross-head Gudgeon Pins.—These pins require to be very accurately fitted, so as to be free from the slightest looseness or

[1] This type for a great many years held its own, and was very largely used, notwithstanding the number of parts and the labour involved in fitting up. The almost universal practice now is to make the guiding surface in one with the frame (as shown in Figs. 636, 640, and 641), which reduces the liability of error in erecting, and also the labour.

[2] The guide bar G is in this case above the cross head. This, of course, is the best position for it in a slipper arrangement for a locomotive, as the greatest pressure is upwards when the engine is running forward. When running backwards the sliding surface taking the pressure is smaller (as with all slipper cross heads), but, in this case, not very much smaller.

[3] The proportions shown are mainly those given by *Unwin*.

[4] The shoes of all these cross heads must be so fitted that they are easily taken down, and there is no possibility of their working loose. Wear takes place after a time, and this is usually taken up by fitting thin strips of Muntz metal between the slipper and the body of the head.

STATIONARY AND MARINE CROSS HEADS.

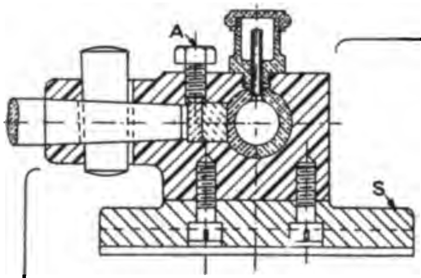


FIG. 654.

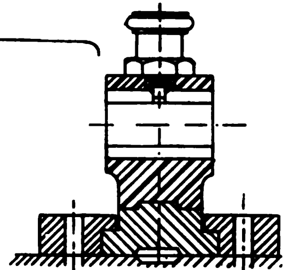


FIG. 655.

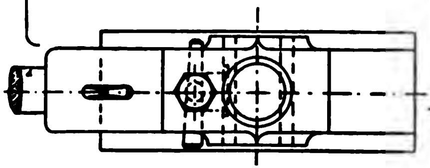


FIG. 656.

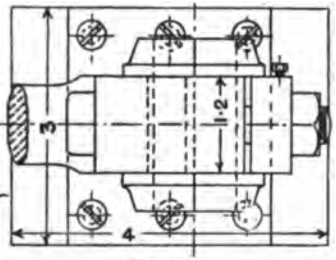


FIG. 657.

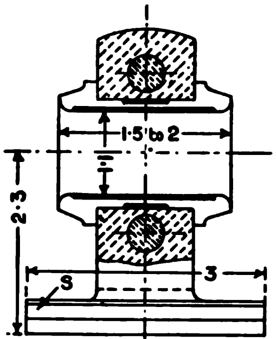


FIG. 658.

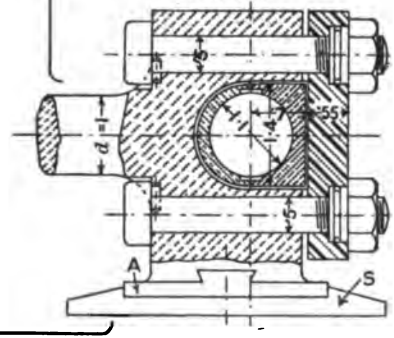


FIG. 659.

shake, and to be held or secured in such a way that they cannot rotate about their axes. Perhaps the most simple and easy way to satisfy these conditions is to make the pin parallel, and a snug fit at its ends, fitting a feather or feathers F, Fig. 673, to it to prevent any movement about the axis. Fig. 674 is a somewhat similar arrangement, a shoulder[1] S being used instead of the head H of the other, and a pin P is sometimes used instead of the feather. Coning the ends, as in Fig. 677, is a very popular expedient, but in practice it is most difficult to get an exact fit at both ends; however, by slightly increasing the diameter of the large end, as in Fig. 675, the two conical ends are parts of the *same* cone, as shown by the dotted lines, and this greatly facilitates the machine work both on pin and head. Fig. 676 shows a pin for a marine cross head; it is forced into the forked end of the connecting rod by hydraulic pressure, or shrunk into it while the fork is hot; usually the pin is further secured in the fork by a strong set-screw, as shown. Such gudgeon pins are made hollow if weight is of importance. As will be noticed in Fig. 677, a snug S may be used on the pin instead of a feather, and the gudgeon pin held in position by the plate A pressed on the shoulder of the pin by the three screws. A simple and effective arrangement of fixing, which allows the gudgeon or cross-head pin to be readily withdrawn, is shown in Fig. 678; it was devised by Messrs. Bollinckx, of Brussels, and is used on their famous engines.

248. Cross Head for Horizontal Engine (Drawing Exercise).—The three views, Figs. 679, 680, and 681, show a cast-iron cross head for a horizontal steam engine. Some particulars of this type are given in connection with Figs. 640 and 641.

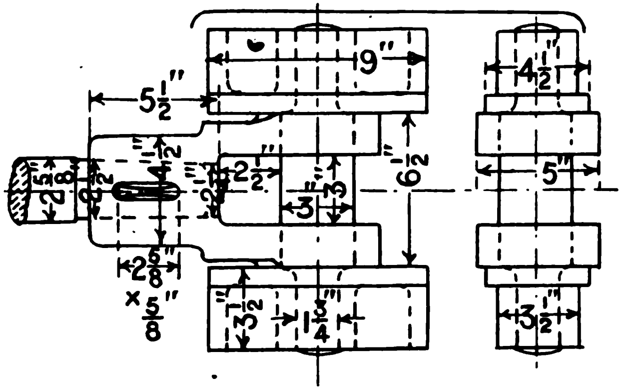
Instructions.—Draw the views shown, completing the section EF. Draw also an end elevation as seen when looking at the end of the rod. **Scale 6 ins. = 1 ft.**

249. Cross Head for Marine Engine (Drawing Exercise).—Fig. 682 shows in detail the separate parts of a marine engine cross head.

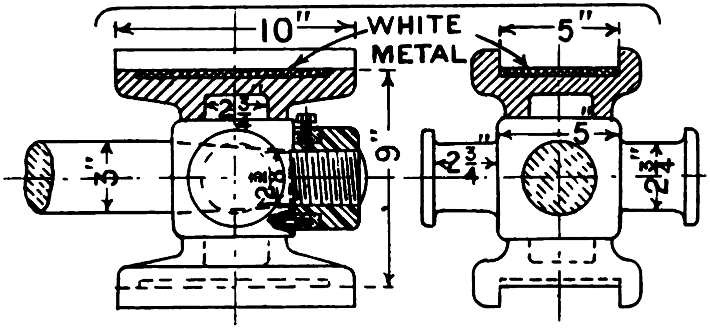
Instructions.—Draw three views of the assembled cross head. Scale quarter full size.

[1] These require to be *well filleted*, as cross-head pin failures with about half Wohler's value of the stress, have to be attributed to sharp shoulders. Refer to Dr. Stanton's experiments, *Proceedings Inst. C.E.*, vol. clxvi.

TYPES OF LOCOMOTIVE CROSS HEADS.



FIGS. 660, 661.—Four-bar type (Drawing Exercise).



FIGS. 662, 663.—Two-bar type (Drawing Exercise).

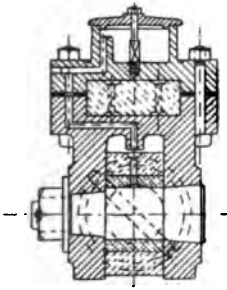


FIG. 664.

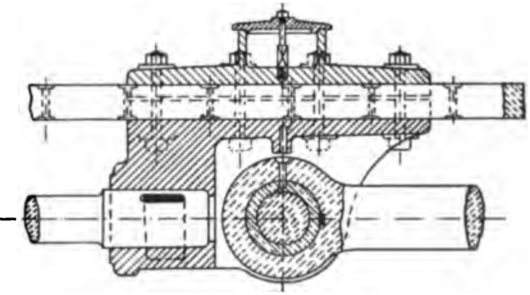


FIG. 665.—Adams's cross head. One-bar type.

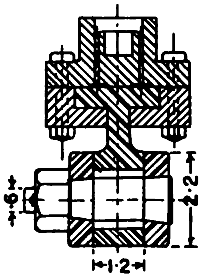


FIG. 666.

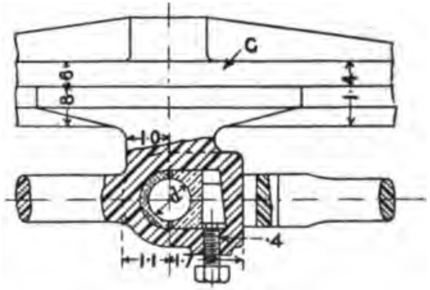


FIG. 667.

Instructions.—Work to the dimensions upon the figure, which is not to scale. And—
(*a*) Draw the cross head in elevation.
(*b*) Draw the cross head as seen looking towards the cap, the top to be a half end elevation, the lower half a vertical section, taken through the centre of the pin. **Scale 3 inches = 1 foot.**
You are *not* to draw the parts separated, as shown, but put them together in their proper positions.
Section line the various parts for suitable materials.

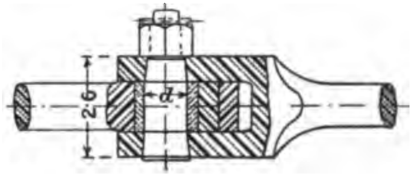
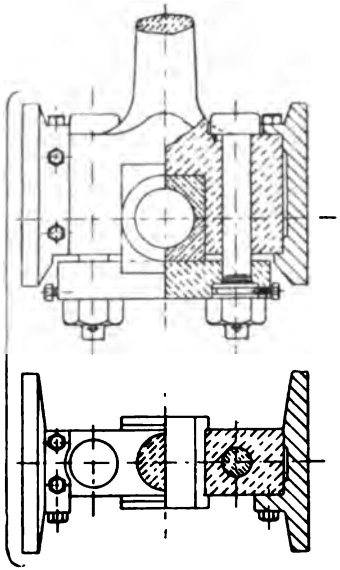
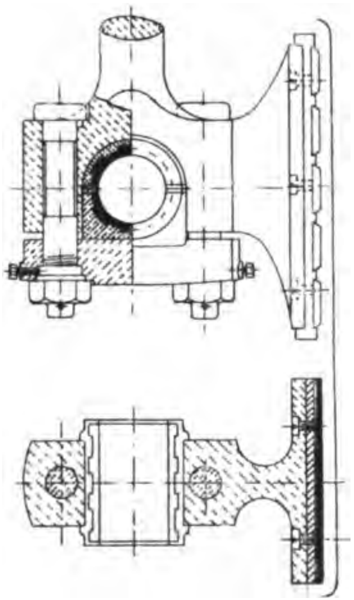


FIG. 668.—Stroudley's cross head. Slipper type.

TYPES OF MARINE CROSS HEADS.



FIGS. 669, 670.



FIGS. 671, 672.

TYPES OF CROSS-HEAD GUDGEON PINS.

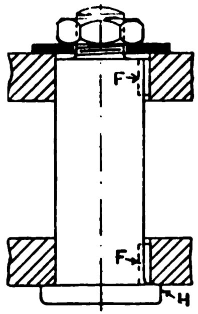


FIG. 673.

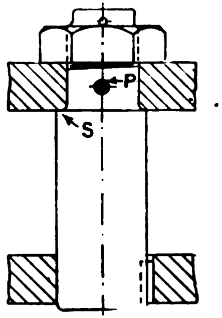


FIG. 674.

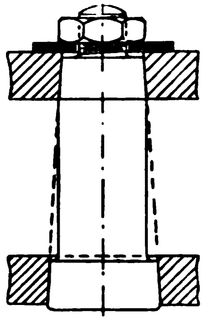


FIG. 675.

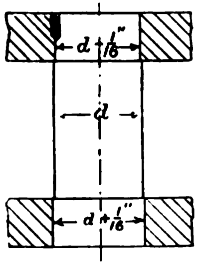


FIG. 676.

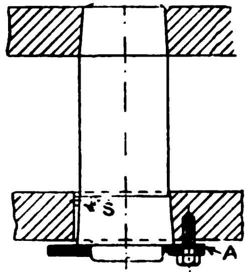


FIG. 677.

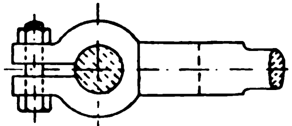


FIG. 678.—Bollinckx.

EXERCISES.

DESIGN, ETC.

1. A connecting rod is five times the length of the crank, and the pressure on the piston is 20,000 lbs. Find, by the triangle of forces, what is the pressure on the cross-head guide when the crank has moved through 45°. And what is the pressure when the crank and connecting rod are at right angles to each other.

2. A gudgeon pin has a diameter of 3″ and a length of 4″, and the journal is loaded to 1100 per sq. inch. What skin stress does this correspond to, and what pressure per sq. inch on the journal?

3. A piston has a diameter of 30″, and when the connecting rod is inclined 15° the pressure on the piston is 80 lbs. per sq. inch. What would be the thrust of the connecting rod on the gudgeon at that instant?

Drawing.

4. Make working drawings of the cross head, Figs. 660 and 661. Scale half-size.

5. Set out the locomotive cross head shown in Figs. 662 and 663. Scale 6″ = 1′.

3. Draw the three views of the cross head in Figs. 637 to 639, full size.

Sketching.

7. Make a sketch of cast-iron cross head suitable for a stationary engine (Figs. 637 and 639).

8. Make a sketch of Stroudley's locomotive cross head (Figs. 666 to 668).

9. Sketch a slipper cross head, marine type, arranged to receive the gudgeon pin of a forked connecting rod (Figs. 671 and 672).

10. Show by sketches three ways of fixing gudgeon pins to cross heads, and state which one you prefer, and why.

CROSS HEAD FOR HORIZONTAL STEAM ENGINE (Drawing Exercise).

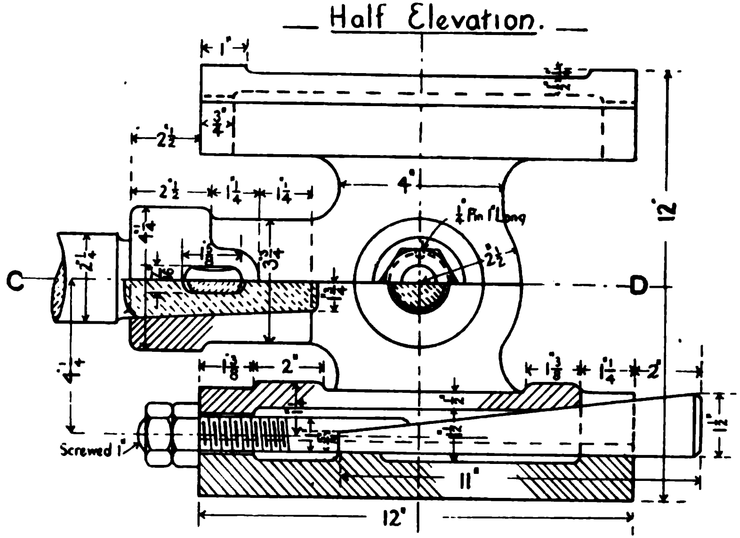


Fig. 679.—Sectional elevation on line AB.

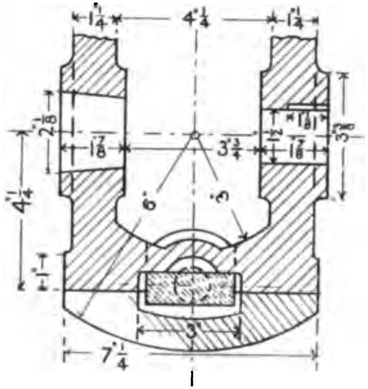


Fig. 681.—Section on line EF.

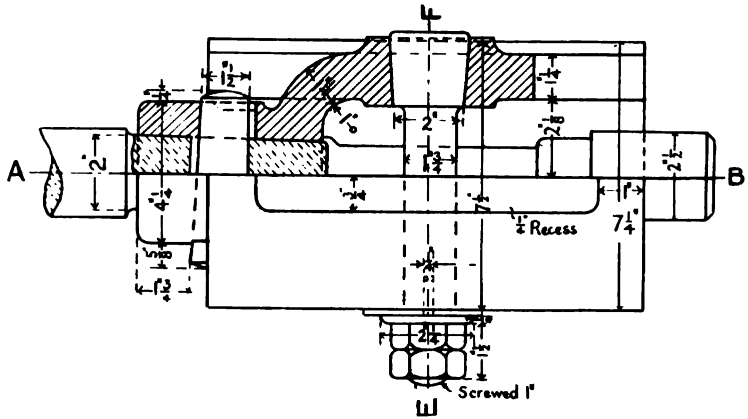


Fig. 680.—Sectional plan on line CD.

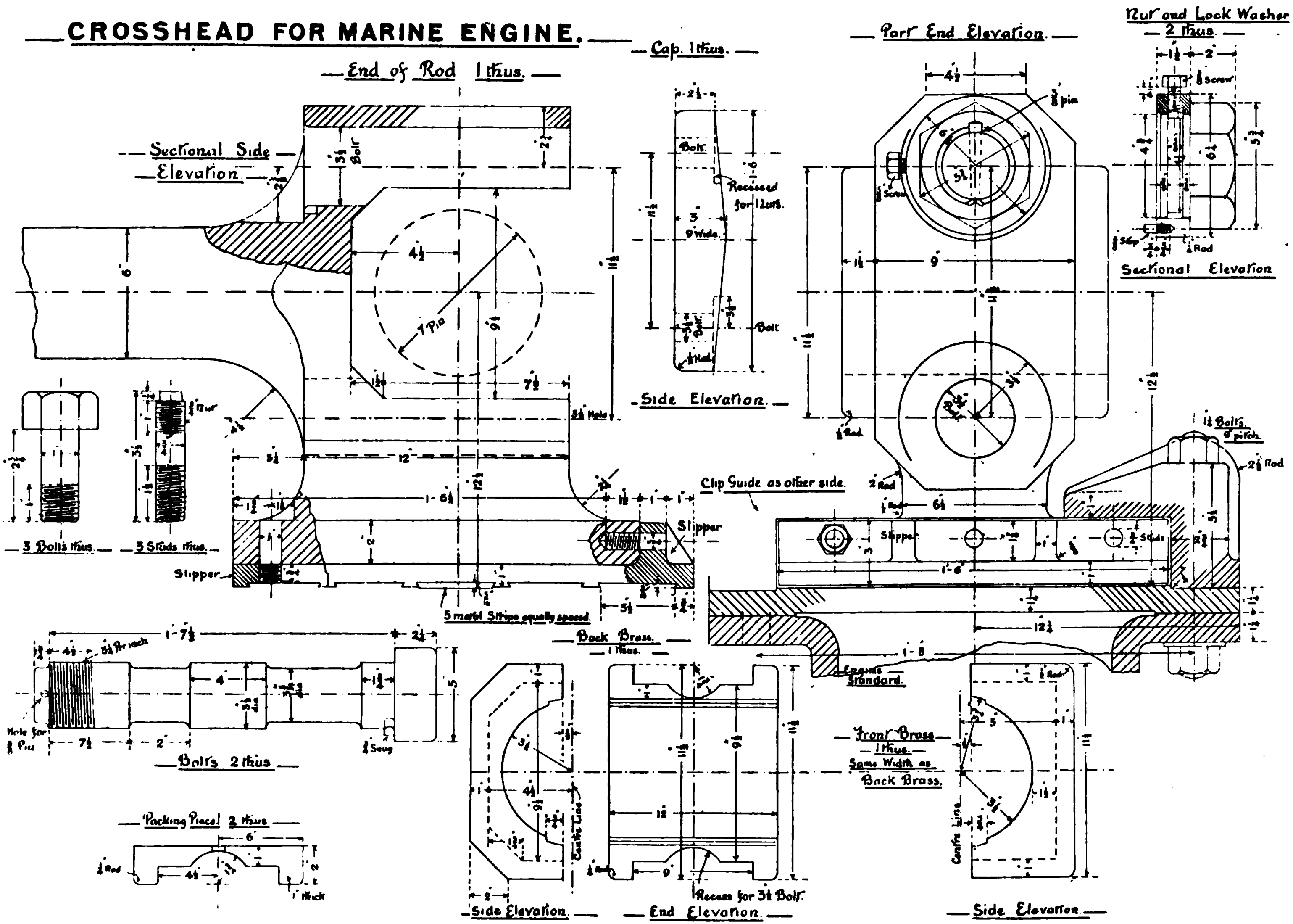


Fig. 682.—Drawing Exercise. For instructions, refer to Art. 249, p. 202.

CHAPTER XXIII

CONNECTING RODS

250. Length of Connecting Rod.—The length of a connecting rod, measured from the centre of the crank pin to the centre of the gudgeon or cross-head pin, varies from about 3·5 to about 8 (or in extreme cases even 9) times the length of the crank, according to the type of the engine; a ratio of 6 to 1 being very generally used for stationary (and often for locomotive) engines. If we had only to consider the rod in fixing this ratio, in a given case, we should make the latter as short as practicable, because the rods acting as struts, must obviously (for a given load on the piston) be larger in diameter as their lengths increase. But we have seen (Art. 244) that the longer the rod is in relation to the crank, the smaller will be the pressure on the cross-head guides, and there is the further important advantage of a more uniform motion of the piston, the longer the connecting rod.[1] However, in practice there are conditions which restrict the length. Thus, the restricted height of marine engines rarely allows of a larger ratio than 4·5 to 1, a ratio of 4 to 1 being more often used, whilst, on the other hand, in locomotive practice, the ratio is generally about 6 to 1, as we have seen.

251. Strength of Connecting Rod.—In addition to the alternate compressive and tensile stresses[2] there is a transverse force applied to the body of the connecting rod due to the oscillation of the crank-pin end. This last factor *is important in small and very high-speed engines,* but not in large ones running at a moderate speed. The diameter of connecting rods for large stationary engines, for the sake of *stiffeners,* is always made greater than would be necessary from a consideration of its ultimate strength as a strut, or its transverse strength against bending. Generally, the largest diameter is such that the mean stress there is 800 or 900 lbs. per sq. inch, and the mean direct stress at the smallest diameter is, as a rule, 1560 to 1600 lbs. per sq. inch.

In **locomotive practice**, an old rule was to make the diameter of the ends about 0·16 × *the diameter of the cylinder*, and that of the centre of the rod 0·21 × *diameter of cylinder;* the former dimension, it will be noticed, is the same as we have given for locomotive piston rods. Molesworth gives the rule, *diameter of connecting rod* $= 0{\cdot}021\mathrm{D}\sqrt{p}$ for iron, and $d = 0{\cdot}018\mathrm{D}\sqrt{p}$ for steel, which for steel nearly corresponds to the previous rule, when the pressure = 150 lbs. per sq. inch, for then $d = 0{\cdot}22\mathrm{D}$.

And again, in **Marine practice**, the diameter d of the connecting rod just below the fork (Fig. 695) is generally made equal to the diameter of the piston rod. Then, if the taper of the rod be produced to the axis of the cross-head pin, d_1 = about $0{\cdot}75d$. And, if the taper is produced in the opposite direction till it cuts a diameter of the crank (Fig. 695), fairly correct values for the larger end are obtained,[3] if $\mathrm{D}_1 = 0{\cdot}6\mathrm{D}$, where D is the diameter of the crank pin. Usually the part of the *shaft* of the rod between the small

[1] The motion of the piston becomes harmonic when the length of the connecting rod is infinite. See Goodman's "Mechanics of Engineering," pp. 133 and 163.
[2] For the magnitude of these, refer to the author's "Machine Design, etc.," p. 514.
[3] Bauer and Robertson's "Marine Engines and Boilers," p. 194.

end and large end is made with a **straight taper**. In some cases rods are made taper from the cross-head end to the middle, and for the remainder of the length parallel.[1] The tapering of the rod not only makes the change of size less sudden, but it gives greater strength to the middle of the rod where it is required to resist bending when in compression. **Flats** are sometimes planed on the taper shaft near the large end and parallel to the plane of motion; this somewhat reduces the section and weight, but very little decreases the strength to resist bending, as the modulus of the section is very little affected. This principle is carried a step further when the rods are made *rectangular* or I-shape[2] in section, as they often are *for high-speed engines, such as* **petrol engines**, or even locomotives. The rods of beam section have been found to answer well when made of cast steel; the section of these at the small end may be made equal in area to that of the piston rod (if of the same material). And the more economical section allows of the large end being made sensibly smaller in section than it would be if of round section, so we see that *the proportions of connecting rods are to a large extent fixed by the application of* **empirical rules**, which long experience has proved to be satisfactory; however, in cases where there is a departure from ordinary practice in any important respect, it is certainly advisable to check the dimensions of the principal parts of the rod, or at least those of the shaft, by determining the maximum stresses, and we will proceed to indicate how this may be done.

252. Connecting Rods for Internal Combustion Engines are usually fitted with big ends of the marine type, and the little ends for **small gas engines** and **petrol engines** are fitted with solid bushes to engage the gudgeon pin; the bushes being easily renewed when necessary. Connecting rods for these engines are practically always in compression,[3] with respect to gas pressures, *but at certain parts of the strokes they are in tension due to inertia.* And (unlike rods for steam engines) when the ignition is early and the other conditions at their best, *the greatest possible thrust on the rod equals the total pressure on the piston at the commencement of the stroke.*[4] So there is little beyond the determination of maximum thrust in designing these rods further than what we have seen applies to steam-engine rods. Except when the main object of the design is to obtain *the minimum weight in a given case,* then *a very careful and complete analysis should be made, by combining gas pressure and inertia curves,* etc.

For **petrol engines**, connecting rods are usually made very light for the work they do. Dr. Lucke[5] gives the following empirical rule for the limiting values of d, the mid-section diameter of **round rods**, namely, between—

$$d = 0{\cdot}011\mathrm{D}\sqrt{p}, \qquad \text{and } d = 0{\cdot}014\mathrm{D}\sqrt{p} \quad . \; . \; . \; . \; . \quad (41)$$

where D is the diameter of the cylinder, and p the initial pressure per sq. inch (about 250, say).

Or for **plain rectangular rods**, the mean thickness t, for rods varying from 15 to 25 thicknesses in length, is—

$$t = 0{\cdot}008\mathrm{D}\sqrt{p} \quad . \; . \; . \; . \; . \quad (42)$$

The width at the piston end is usually $1{\cdot}6t$, and at the crank end $2{\cdot}3t$. A common practice now is to make the connecting rods of I

[1] Long rods with ends equal in size are usually made **barrel shape** with the diameter at the ends = $0{\cdot}75d$ to $0{\cdot}85d$, or the full diameter is about 0·4 the length from the crank end, and the diameter at the crank end is about $0{\cdot}9d$, and at the cross-head end $0{\cdot}8d$. But if short and with equal ends the rods are often parallel.

[2] Some locomotive rods are milled at each side to give this section, which reduces the weight, and thereby the effect of the *inertia forces, so that the rod is really increased in strength by being made lighter.*

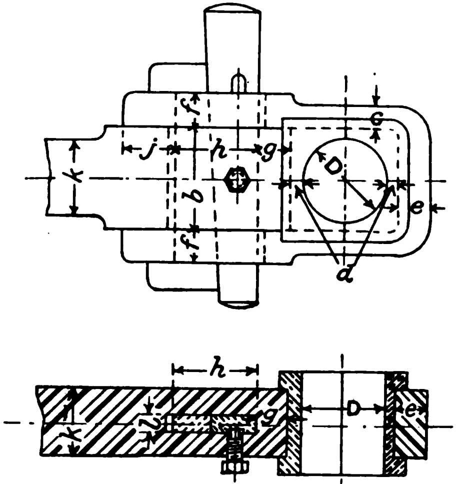
[3] This should be borne in mind in fixing the size of the bolts for the big end.

[4] In explosion engines the maximum pressure occurs about the commencement of the stroke (see the author's "Motors and Motoring," p. 55); but we have seen that the full pressure is often maintained in a steam cylinder till the position of maximum angularity of connecting rod is passed.

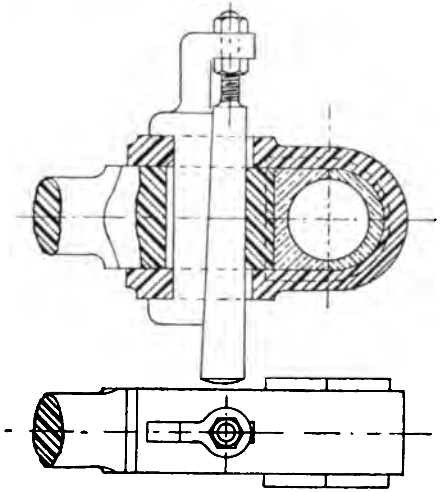
[5] "Gas Engine Design," p. 215.

section (in cast or stamped steel) for reasons explained in the previous article, then the breadth of the flanges may be 1·3 times the t found as above, and the thickness of the web, about $0{\cdot}6t$.

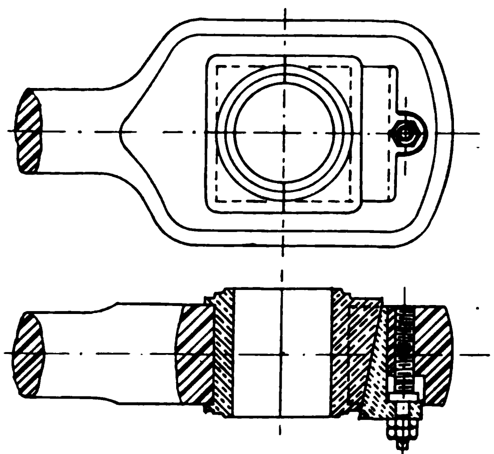
253. Connecting-rod Ends.—We have seen that in any steam engine the crank pin must be larger than the cross head, for reasons that have been discussed; this being so, the end of a connecting rod which forms the bearings for the gudgeon or cross-head pin, and the one for crank pin must be very different in size, in fact they are commonly called **little ends** and **big ends**, respectively. Much skill and ingenuity have been brought to bear upon the design of these ends with the object of making them as perfect as possible, with the result that a large number of different forms have been produced, each one with some feature which entitles it to attention; but for some years there has been a growing tendency to favour two or three familiar types, and these may be regarded on the whole as the survival of the fittest. Hence we find these types, with slight variations in detail, in general use in modern practice; so some representative ones, and others that are interesting variations, have been selected by the author to put before the student. Figs. 683 to 686 are examples of the **plain strap end**, which has been used more largely than any other in stationary engines; it makes an excellent job, but is costly when well fitted. It will be seen that the rod itself is finished with a rectangular end, through which a slot is cut to receive a gib and cotter. A brass with a square back is fitted to the end of the rod, and a corresponding one, with either a round back (as in Fig. 685) or flat back (Fig. 683), fitted



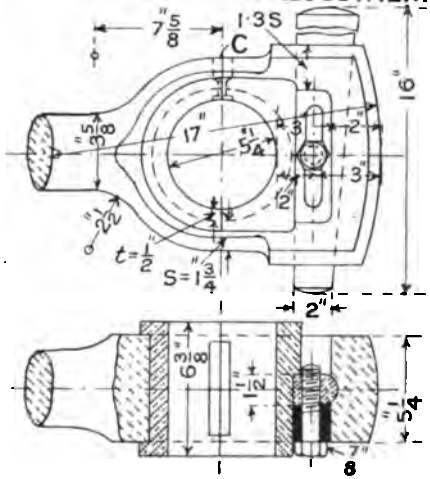
FIGS. 683 & 684. PLAIN STRAP PATTERN.



FIGS. 685 & 686. PLAIN STRAP PATTERN WITH SCREW ADJUSTMENT.



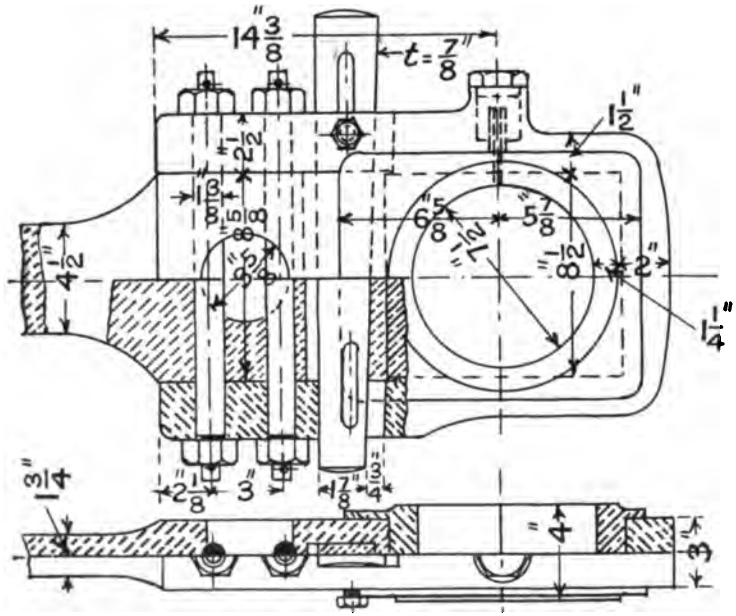
FIGS. 687 & 688. SOLID END WITH SIDE ADJUSTMENT



FIGS. 689 & 690.

opposite, both being provided with flanges on all sides, and made thicker at the bed and crown than at the joint, to provide for wear and also for stiffness. A wrought-iron strap is fitted round both brasses, and cotter slots are cut to match those in the rod. The strap is made thicker at the crown than at the sides to strengthen it against bending due to the ordinary load and to the hammering action which sometimes occurs. The thickness of the strap is also usually increased, as shown, to give a good bearing surface,[1] for the gib, and to compensate for the material removed in the slot. In Fig. 683, the cotter is held in position by a set-screw, but for important jobs it is usual to provide a jib screw arrangement for fine adjustment,[2] as shown in Fig. 685.

CONNECTING-ROD BIG END, LOCOMOTIVE TYPE.



FIGS. 691, 692 (Drawing Exercise).

The following **proportions for a strap connecting-rod end** are given by Molesworth, the *unit* being D, the diameter of the crank pin, and they refer to Figs. 683 and 684 :—

$a = 0{\cdot}1D + 0{\cdot}15$	$e = 0{\cdot}33D + 0{\cdot}06$	$h = D$
$b = D + 1{\cdot}4a$	$f = 0{\cdot}37D + 0{\cdot}12$	$j = 0{\cdot}5D + 0{\cdot}4$
$c = 0{\cdot}3D + 0{\cdot}06$	$g = 0{\cdot}35D + 0{\cdot}12$	$k = 0{\cdot}9D$
Taper of cotter 1 in 16		$t = 0{\cdot}2D + 0{\cdot}06$

In some cases of strap ends a safety bolt is also passed through the whole end, parallel to the cotter; a modification of this arrangement being largely used for locomotive rods. Figs. 691 and 692 show an example of a big end dimensioned for an 18″ **locomotive** cylinder, and steam pressure[3] of 150 lbs. per sq. inch. The **bolts** have $\frac{1}{8}$″ **taper** on them to facilitate disengaging; the mean diameter being shown. A modified form of the same arrangement is used for the little end, as shown[4] in Figs. 703 and 704. When well fitted and in good proportion this type makes an excellent but expensive job; having practically the advantage of a solid end, it can be taken apart for use with double webbed cranks, and can be used with great safety at high speeds.

Examples of **solid ends** are shown in Figs. 687, 689, 695, 705, and 707. This type of ends has steadily grown in favour since it appeared on the Allen engine some forty years ago. A large solid end is forged on the rod, through which an eye of

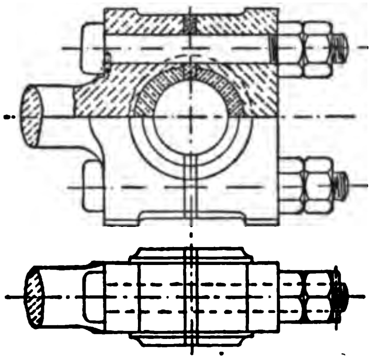
[1] The compressive stress f_c should not exceed about 2·5 tons per sq. inch.

[2] If both ends of a rod were alike (Fig. 685), the latter would be **shortened in cottering up**. Sometimes they are arranged so that one end shortens and the other lengthens. For instance, Figs. 689 and 691.

[3] For this pressure, and taking the thrust on the rod as equal to the pressure on the piston, the principal parts are subjected to the following stresses; strap at thinnest part 4235, strap at bolts 4500, shear stress in bolts 7050, in large end of the rod itself 4840 lbs. per sq. inch.

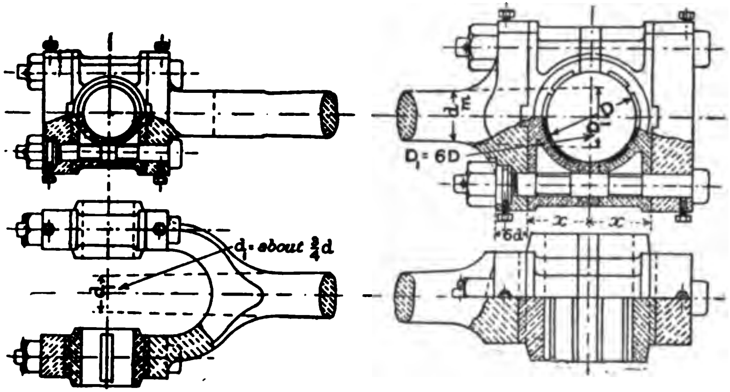
[4] This is a "little end" of a rod for a 17″ locomotive cylinder, pressure 150.

rectangular form is cut, leaving rounded corners and sometimes a rounded end, as in Figs. 689 and 701; and brasses are fitted with flanges where possible, but it is a slight defect that these cannot extend all round. Different arrangements are made for the adjustment of the brasses, as shown in the figures, but when a die D is used, as in Figs. 701 and 707, it should not have a length less than two-thirds that of the opening, and its corners should be left square so as to obtain the largest bearing possible.[1] The **solid end type** can obviously be only used with crank pins on discs or crank arms, but it probably has the advantage of being **lighter than other forms** of ends, and is found to be very free from break-downs and other troubles. One of the simplest and least expensive rod ends is the **Marine type,** shown in a simple form in Figs. 693 and 694, and in a more complete form, as arranged for powerful marine engines, in Figs. 695 and 696. In the latter the shaft is provided at its ends with flat tables or palms, against which the inner brasses are bedded, the outer ones meet these, and wrought-iron plates or caps are fitted outside to give a good support to the brasses. Two strong bolts (usually made with plus threads, and collars where required, as shown) pass through the hole for each pair of brasses, the nuts are secured by set-screws or safety rings, with or without the addition of lock nuts. As the **brasses** are more severely strained than any others on the engine, they should be **made of the finest material,** and designed with the greatest care. The brasses for this type become very heavy and expensive for large size crank pins, and

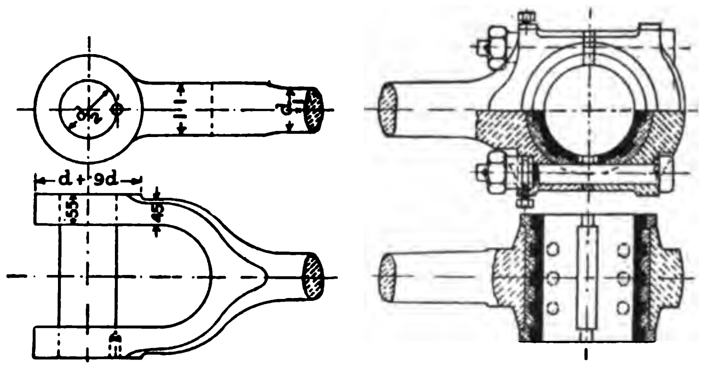


Figs. 693, 694.—Marine type. For small engine.

MARINE CONNECTING RODS.



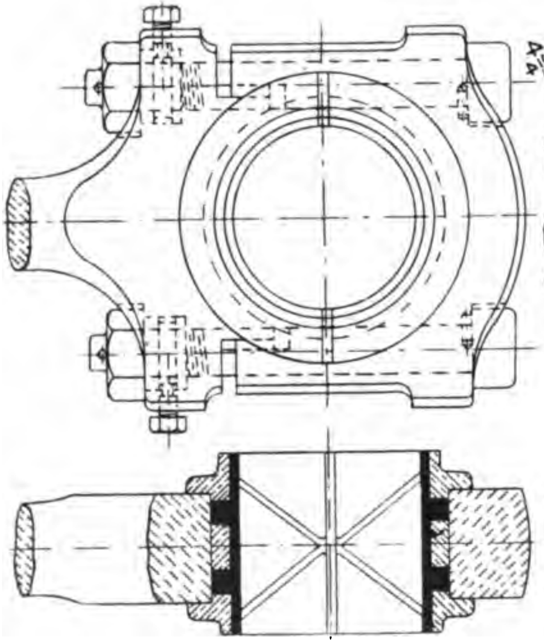
Figs. 695, 696.—Marine type.



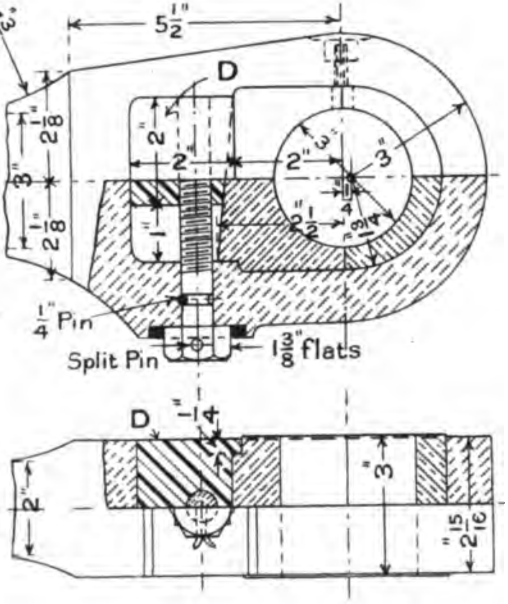
Figs. 697, 698.—A more expensive marine type.

[1] In designing ends like Fig. 689, the cotters must be made thick enough to take the load on the rod without overloading the brass where it beds, as experience proves that hard gun-metal should not be loaded with more than 0·75 ton per sq. inch in still contact with iron.

costly to renew when worn, besides being liable to crack through the crown. So, to avoid the use of so much brass, some engineers prefer the **brasses** to only **act as bushes or liners**, as in Figs. 697 and 698; this avoids the use of much expensive metal and gives a good solid bed to the brasses;[1] it also relieves the bolts of all strain except tension. The brasses are kept from turning either by a brass distance piece held between the cap and rod, and projecting between the brasses, or by projections from the back of the brasses fitting recesses in the cap and rod end. Crank-pin brasses are usually fitted with **white metal**, as it is on the whole a better metal than bronze for the rubbing surface of this part, but it is important that the white metal should project beyond the brass so that it alone is in contact with the pin; this being so, the **brasses** are sometimes made of **cast steel or cast iron.** Figs. 699 and 700 show a variation of the type we have just described; it is a Continental form, with deep cap, and can be made a fine job, but it is obviously more expensive to manufacture.



FIGS. 699, 700.—Heavy marine type. Continental form.



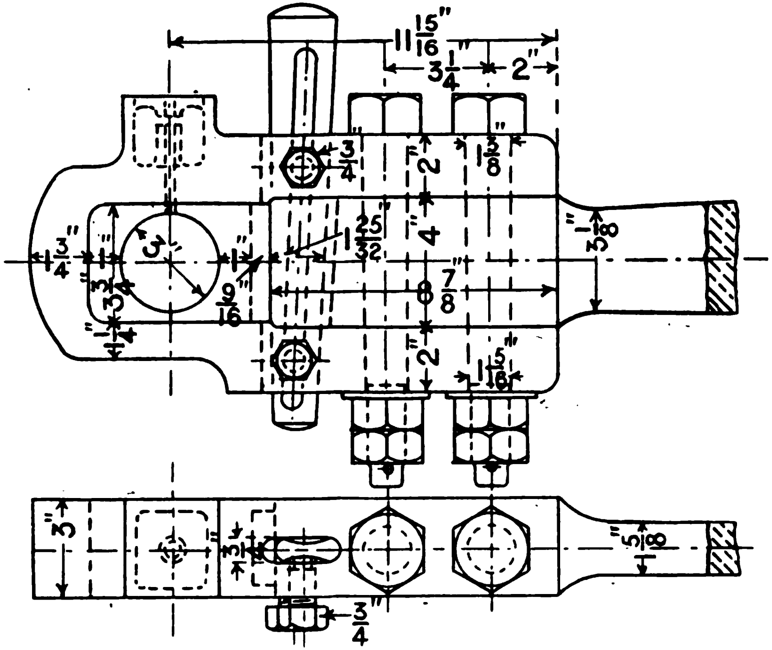
FIGS. 701, 702.—Little end, stationary type (Drawing Exercise).

Four different patterns of **little ends** are shown in Figs. 701 to 708; we have referred to some of these, and it only remains to call attention to Fig. 705, which shows how Messrs. *Musgrave*[2] & Co. have overcome a defect in the ordinary gudgeon brasses; ordinarily the cross-head pin wears only on two sides, leaving the pin approximately oval, when it becomes *impossible to prevent* **back lash**, but by cutting **a flat on each side of the pin**, and cutting away the brasses to correspond,

[1] It is usual to forge the head of the rod solid when it is of this form. The whole end is then turned, and the hole for the brasses is then roughly bored or slotted out, the head is then slotted through or parted in the lathe so as to cut off the cap; the bolts are then fitted and the cap bolted close to the rod, and the hole bored out to receive the brasses. The end in Fig. 693 is made in the same way. The trouble with all **brasses** is that *they tend to* **close on the journal** *after they have been heated*, and this is more pronounced when the brass is in the form of a bush on a rigid bed, as in these cases.

[2] Mr. Halpin's name is also mentioned in connection with this arrangement, but the author is not aware who has the prior claim.

as shown, the brasses in vibrating through a small angle overrun the edges of the flats on the pin, and shoulders are not formed by wear. Furthermore, the lubricant more easily reaches the wearing surfaces.



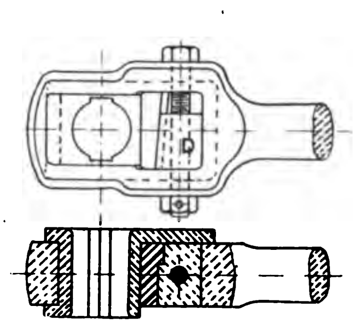
FIGS. 703, 704.—Little end, locomotive type (Drawing Exercise).

254. Connecting-rod End for Marine Engine (Drawing Exercise).—Fig. 709.

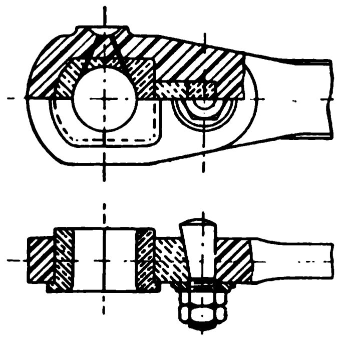
Instructions.—Draw the sectional elevation and sectional plan, also an end elevation as seen when looking in the direction of arrow A. Take dimensions for the smaller details *from sketches* given. Section each part to indicate material used. **Scale one half full size.**

255. Connecting-rod End, Forked Type (Drawing Exercise).—Figs. 710 and 711 show two views of the little end (forked type) of a connecting rod for a vertical engine. This example was set at the C. and G. Examination in Mechanical Engi-

CONNECTING-ROD LITTLE ENDS.



FIGS. 705, 706.—Solid end, with Musgrave brasses.



FIGS. 707, 708.—Solid end, with side adjustment.

neering in 1907, Part II., Ordinary Grade. Students were given the following :—

Instructions.—The figures show two incomplete views of the little end of a connecting rod for a vertical steam engine. Draw to a scale of 3″ to 1′ :—
(1) A half sectional elevation, the upper half being in elevation and the lower half in section, on the plane AB.
(2) In place of the plan, a section on the plane CD of one-half of the rod end.
(3) A section on the plane EF of one side of the rod end, and a complete end elevation of the other side.
Any omitted detail must be added, together with a suitable oil catcher.

256. Petrol Engine Connecting Rod (Drawing Exercise).—Figs. 712 and 713.

Instructions.—Draw the connecting rod and piston, assembling all the parts shown separately on the figure as when in use, showing front sectional elevation, side elevation, and a plan. **Scale full size.**

CONNECTING-ROD END FOR MARINE ENGINE (Drawing Exercise).

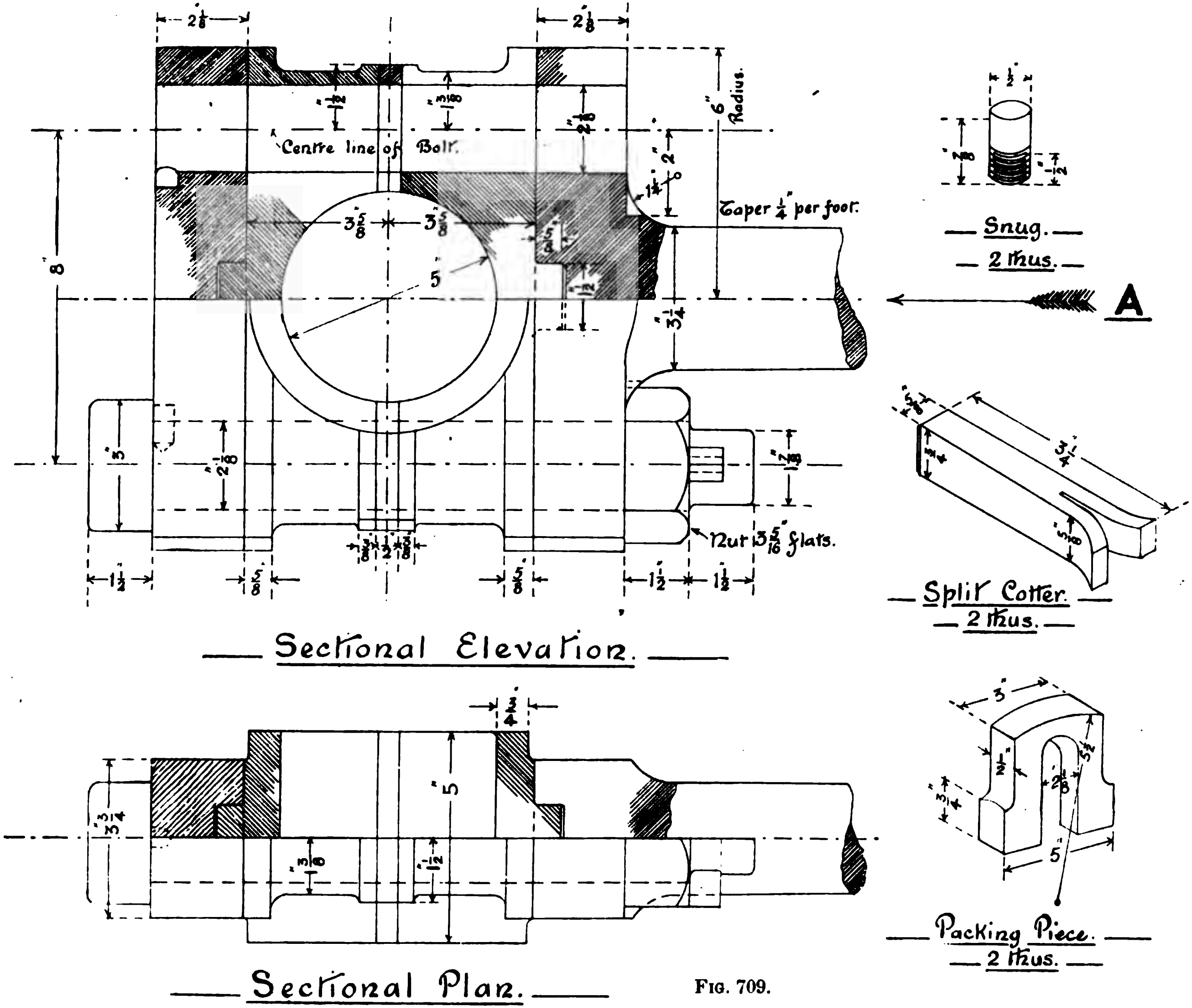
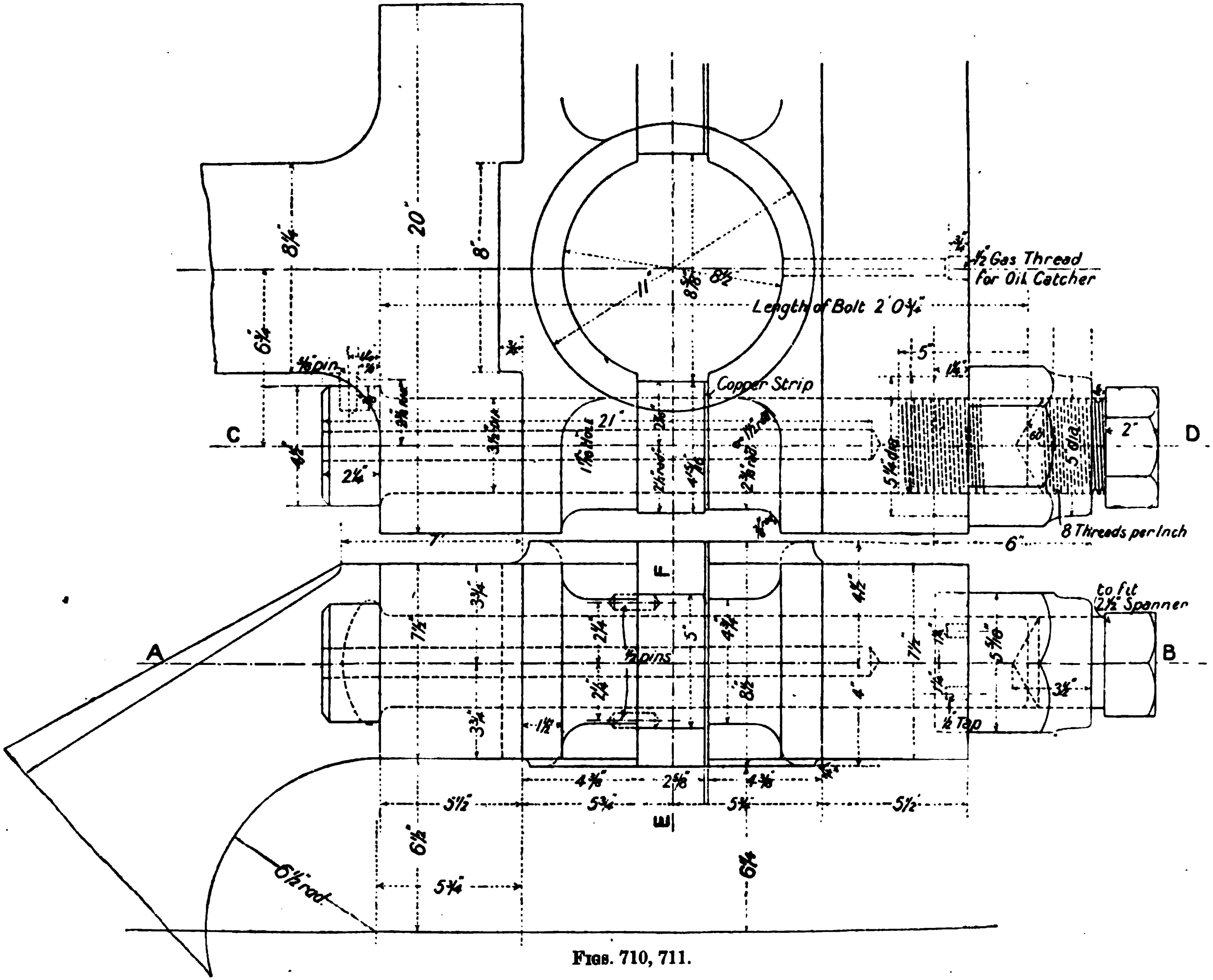


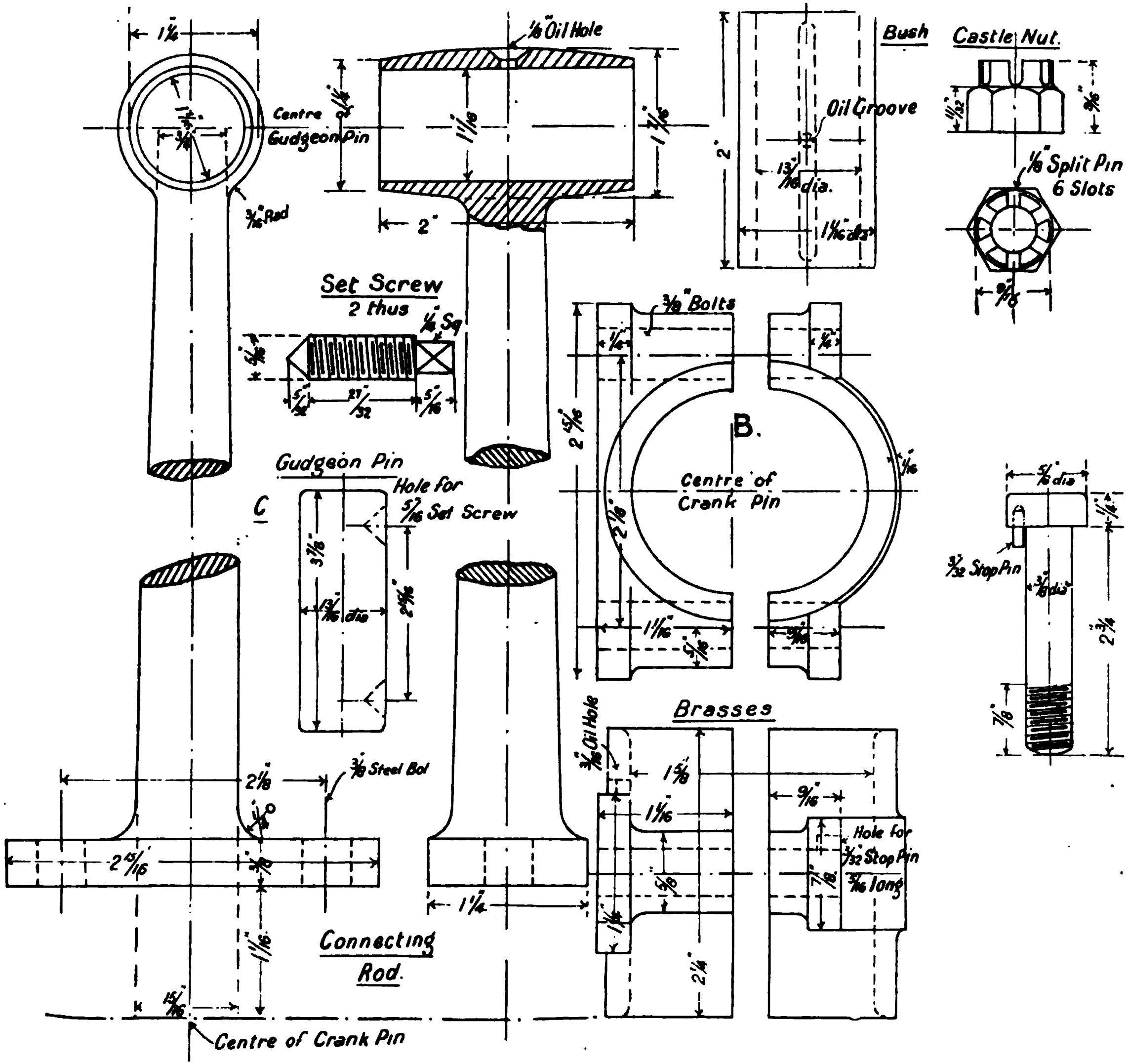
FIG. 709.

CONNECTING-ROD END FOR VERTICAL STEAM ENGINE (Drawing Exercise).



FIGS. 710, 711.

PETROL ENGINE CONNECTING ROD (Drawing Exercise).



FIGS. 712, 713.

257. Connecting Rod of Petrol Engine (Joist Type).[1]—Figs. 714 and 715 show a lighter form of rod, which is now much used.

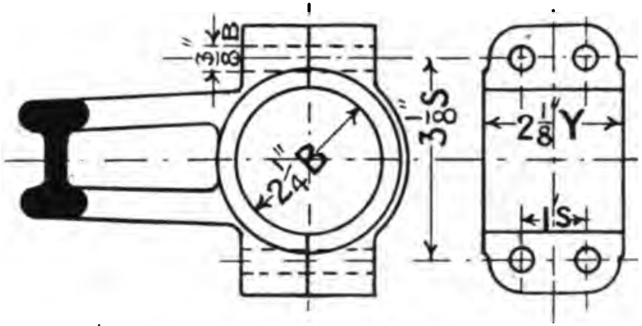
258. Connecting-rod Brasses.—The total load and the permissible pressure per sq. inch (see Table 14) determine the dimensions of the brasses. When the crank pins are of large diameter the brasses are made, as we have seen, of bronze, or of cast steel with white metal linings, and the distance x from centre to back of brass, Fig. 695, may be such that for **solid gun-metal or cast steel lined with white metal**—

$$x = 1{\cdot}3\frac{D}{2} \text{ to } 1{\cdot}4\frac{D}{2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (43)$$

And for **gun-metal brasses lined with white metal**—

$$x = 1{\cdot}35\frac{D}{2} \text{ to } 1{\cdot}5\frac{D}{2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (44)$$

the higher values being for the smaller pins in each case.



Figs. 714, 715.—Petrol engine connecting-rod end, joist type.

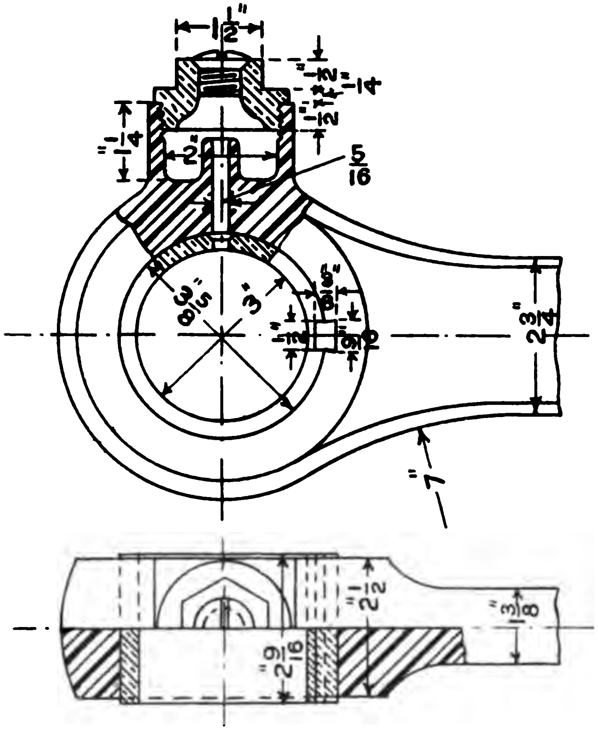
For circular-shaped brasses, Figs. 697 and 698, and other forms shown on the rod ends, in fixing the dimensions, the proportions shown on Figs. 427A to 427S, Art. 176, may be used as a guide, the unit for this purpose being—

$$t = 0{\cdot}08D + 0{\cdot}125'' \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (45)$$

259. Coupling-rod Ends.—The rods used in locomotives to transmit the motion of the crank axle to another or other axles coupling them together, so that a larger proportion of the whole weight of the engine may be available to increase the adhesion on the rails, are called *coupling rods;* they are made of steel or wrought iron, and usually of rectangular or I section, and the ends are now made solid; as there is not a great amount of wear, the lubrication being very good, so no adjustment for wear is usually arranged, a solid brass bush, with a key or feather to prevent rotation, being fitted to each end, this being easily replaced by a new one when worn out. An ordinary form of coupling-rod end[2] for a four-wheel coupled engine is shown in Figs. 716 to 718. And Figs. 719 and 720 show how the rods are generally arranged and connected when more than two pairs of wheels are coupled together. They are drawings of a pattern adopted by the M. S. & L. Railway Co. for their goods engines.

[1] Refer to Art. 482, author's "Machine Design, etc."
[2] This example is taken from a past B. of E. Examination paper. Wordsell's Coupling-rod End is illustrated on Sheet 23 of the author's "Elements of Machine Construction and Drawing."

LOCOMOTIVE COUPLING-ROD END.



FIGS. 716, 717.—Type of end for four-wheel coupled engine.

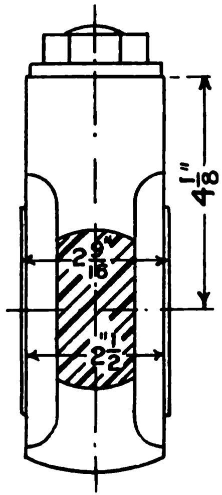


FIG. 718.

EXERCISES.

DRAWING, ETC.

1. Make working drawings of the connecting-rod big end. (Figs. 685 and 686.) Scale full size.

2. Make plan, elevation, and end elevation, in part section of the locomotive connecting-rod big end. Figs. 691 and 692. Scale 6″ = 1′.

3. Set out a strap connecting-rod big end from the proportions given in connection with Figs. 683 and 684; diameter of crank pin, 2″. Full size.

4. Draw the two views of the connecting-rod little end (Figs. 695 and 696), and add an end view. Full size.

5. Draw three views of the locomotive connecting-rod little end (Figs. 703 and 704), half size, and calculate the mean shear stress in bolts; also the tensional stress in the rod, and in the strap at the bolt hole, assuming that the maximum load on the rod is 34,000 lbs.

6. Draw three views of the coupling-rod end (Figs. 716 to 718) partly in section. Full size.

7. Set out three views, partly in section, of the coupling-rod end (Figs. 719 and 720). Full size.

SKETCHING.

8. Make a sketch of a plain strap connecting-rod end, with screw adjustment (Figs. 685 and 686). Why is the strap made thicker at the end and where the brass beds?

9. Make a sketch of a locomotive connecting-rod big end (Figs. 691 and 692), and explain how, if the other end be like Figs. 703 and 704, the true distance between centres is practically maintained when wear occurs.

10. Make a sketch of a *solid end* for a connecting rod. What are the good points of this type, and what feature limits its use?

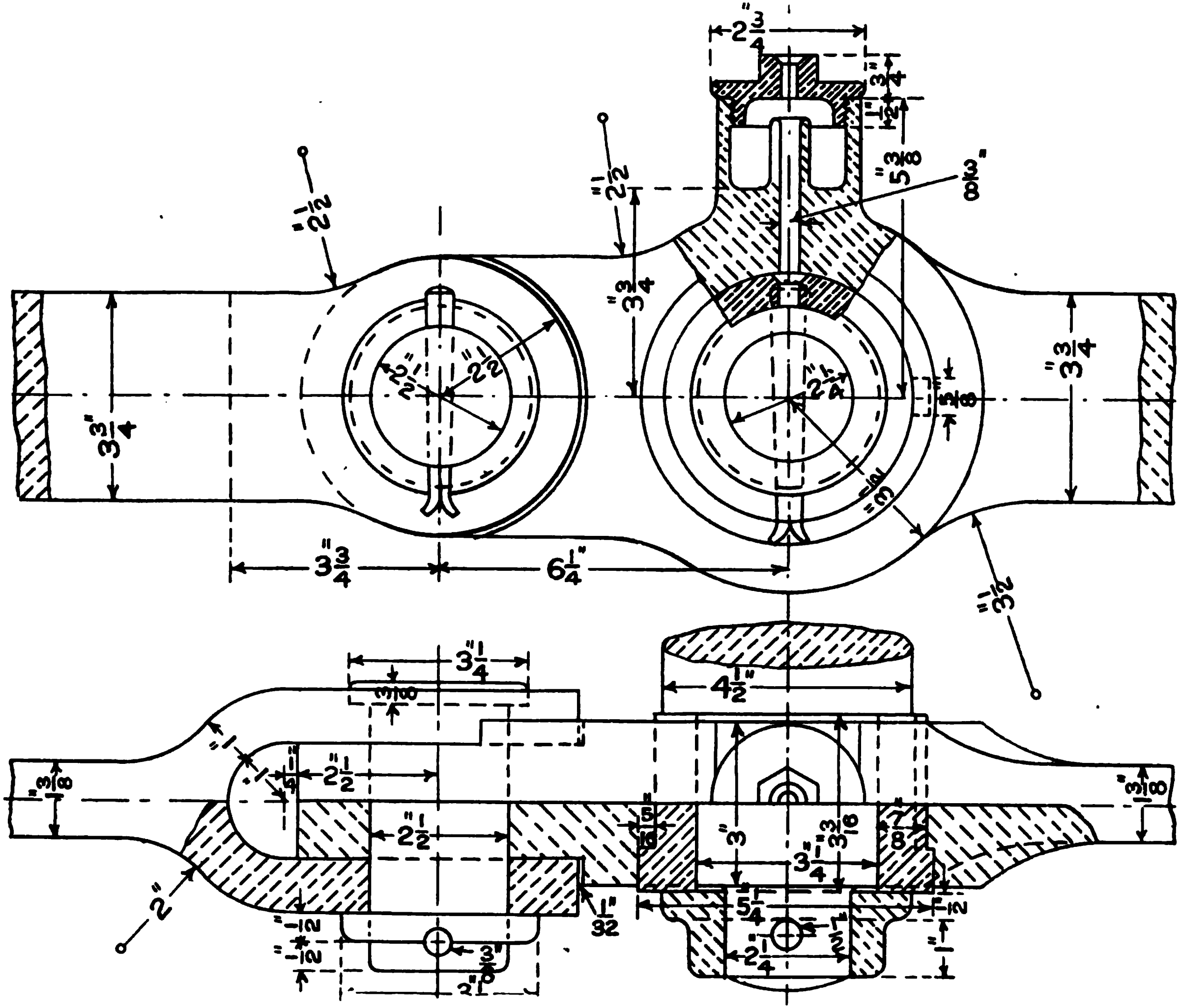
11. Make sketches of two patterns of marine connecting-rod big ends, and explain their relative good features.

12. Sketch two patterns of connecting-rod little ends, and say which you consider makes the best mechanical job, and why.

13. Show by a sketch how you would arrange the ends of a connecting rod so that tightening up for wear at both ends practically keeps the rod centres constant in length.

14. Make a sketch showing Musgrave's or Halpin's brasses and pin for a connecting-rod little end. What advantage is claimed for this arrangement over the ordinary one?

LOCOMOTIVE COUPLING-ROD ENDS AND JOINTS.



FIGS. 719, 720.—Type of ends and joints for rods coupling six or more wheels.

CHAPTER XXIV

MISCELLANEOUS DRAWING EXERCISES

260. Cast-iron Bracket.—The front and end elevation of a cast-iron bracket[1] are shown in Figs. 721 and 722.

Instructions.—Draw these views, and also a plan. Scale full size.

CAST-IRON BRACKET.

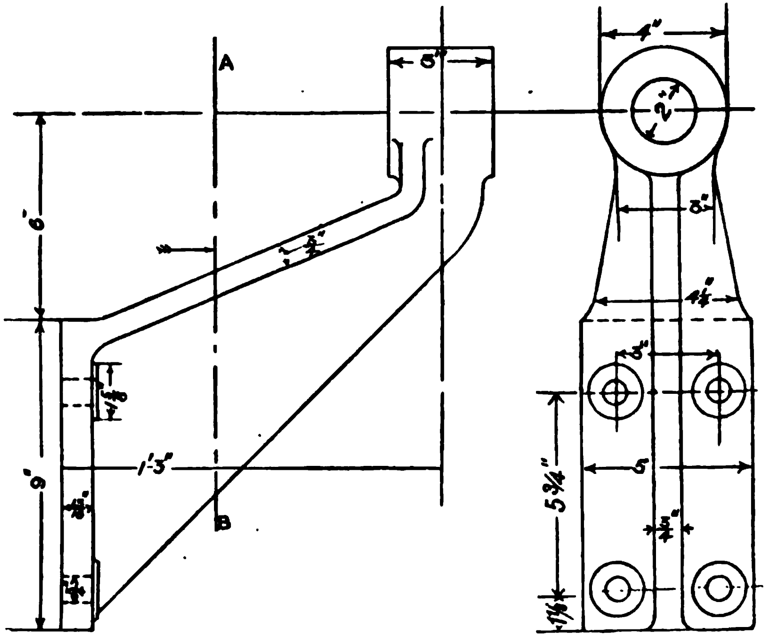


Fig. 721. Fig. 722.

261. The Lathe Bed Bracket, shown by two pictorial views in Figs. 723 and 724, was set as an example in Stage 1 of the B. of E. paper in 1907.

The **instructions** were—Draw full size, inserting dimensions:—
(*a*) An elevation as seen when looking in the direction of the arrow R. Put in the ½" set-screw and ⅝" stud.
(*b*) A sectional elevation on a plane parallel to the face H, and 1" distant therefrom; that is, the section plane is taken through the axis of the ½" set-screw and ⅞" hole.
(*c*) A plan.

N.B.—Do not draw the pictorial view. Dotted lines, representing hidden parts, are not required.

An **adjustable bearing**[2] for bolting to the table of a small boring machine for the support of the boring bar is shown in Figs. 725 to 726A.

Instructions.—Draw a sectional elevation through FG looking in the direction of the arrow H, and make a complete plan. Indicate the screw threads by any conventional method you please.
Scale ¾ full size.
Neither dotted lines nor dimensions need be shown.
N.B.—Take the vertical centre line in the direction of the longer dimension of your drawing paper.

262. Bed Plate and Standard for Vertical Engine.—The bed plate, brasses, and standard, etc., are shown in detail in Fig. 727.

Instructions.—Draw the bed plate with standard, brasses and guide bars in position. Showing plan and front and side elevations. **Scale ¼ full size.**

263. Bed Plate and Brackets for Dynamo.—Figs. 728 to

[1] Pattern-makers were asked to describe the construction of a pattern for this bracket in the Part II. Ordinary Stage, Mechanical Engineering C. G. Paper of 1907.
[2] B. of E. Paper, Stage 2, 1907.

731 show very fully in detail the bed plate, and fully dimensioned views of the commutator and pulley brackets are shown in Figs. 732 to 735.

CAST-IRON LATHE BED BRACKET.

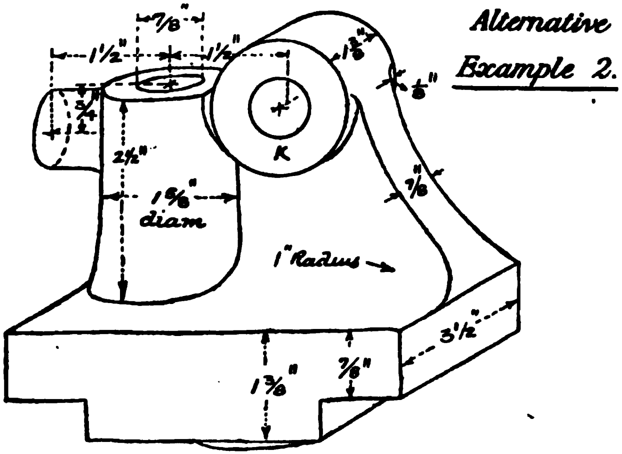


Fig. 723.

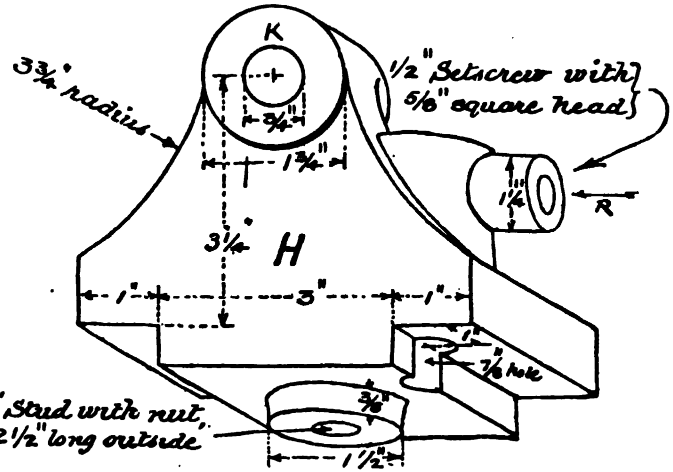


Fig. 724.

Instructions.—Draw the *bed plate* placing the *commutator and pulley brackets* in their *proper positions.* The part to the *left* of the *centre line* to be shown in *elevation* and *plan.* The part to the *right of centre line* as a *section on line* AB. Show *also* a *sectional elevation* on *line* CD, looking towards the *left.* *Indicate* by *edging the surface thus* ////////////// the *parts to be machined.* Each view is to be properly projected; and parts shown incompletely are to be completed and material indicated by systematic section lining. The dimensions need not be shown. **Scale half full size.**

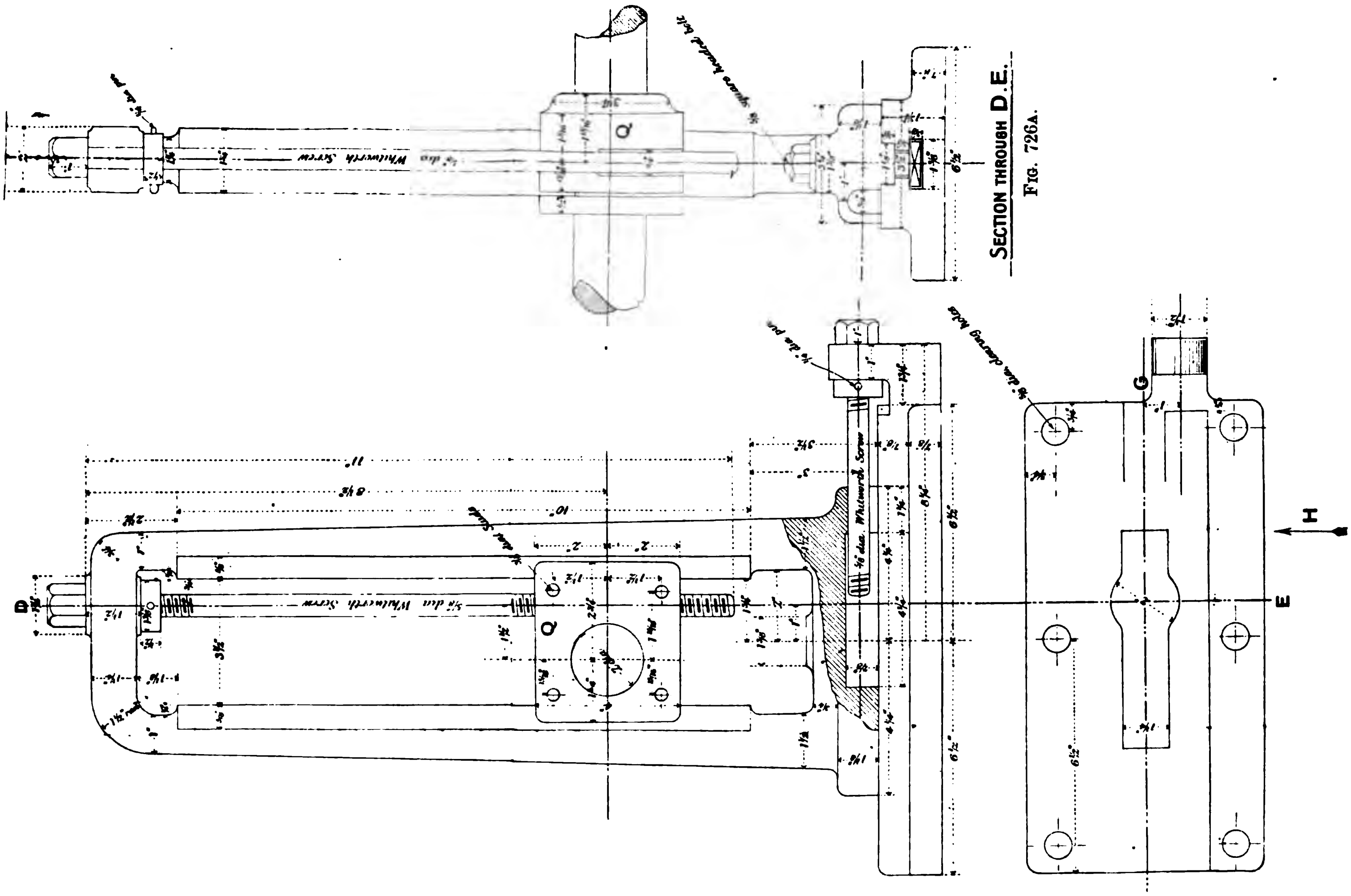
264. Expansion Slide Valve[1] **for Compound Vertical Engine.**—Longitudinal and transverse sections in elevation are shown in Figs. 736 and 737. They are projected one from the other in the same line, but have been placed one below the other for convenience in printing.

Instructions.—Draw the sections of the main and expansion valves on lines AA and CC, also a plan of the valves with the cover Z removed. **Scale 4½" to the foot,** or 3" **to the foot.** The former will make the best drawing if a drawing board large enough is available.

265. Some Details of a 4 Horse-power Single Cylinder Petrol Engine.—In Fig. 738 are shown detailed views of the crank case, crank shaft, piston, and connecting rod, etc., of a small petrol engine.

[1] For information as to the force required to move a slide valve, refer to the author's "Machine Design, etc.," p. 536.

ADJUSTABLE BEARING FOR BORING MACHINE.



Section through D.E.

Fig. 726a.

Figs. 725, 726.

BED PLATE AND STANDARD FOR VERTICAL STEAM ENGINE (Drawing Exercise).

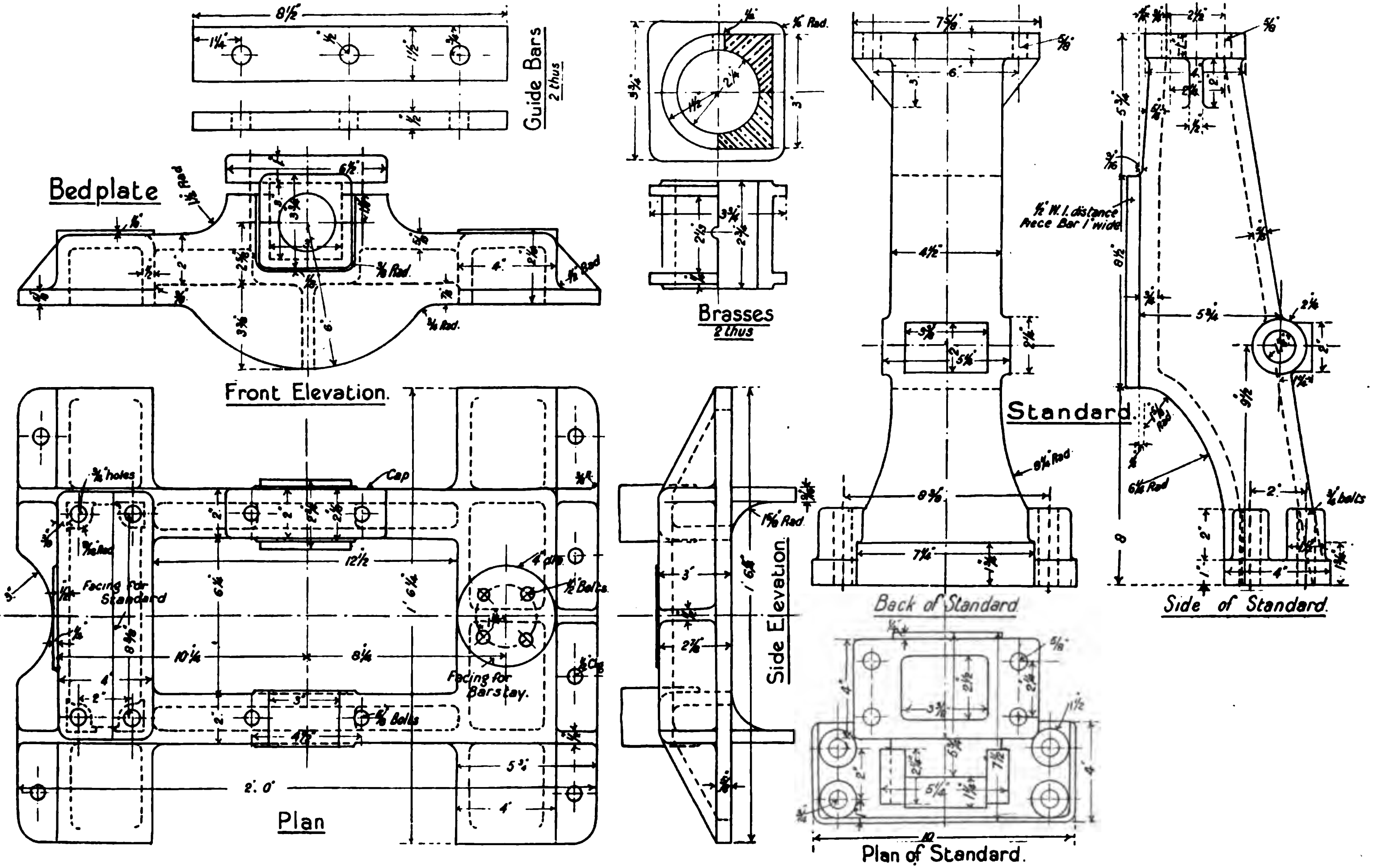
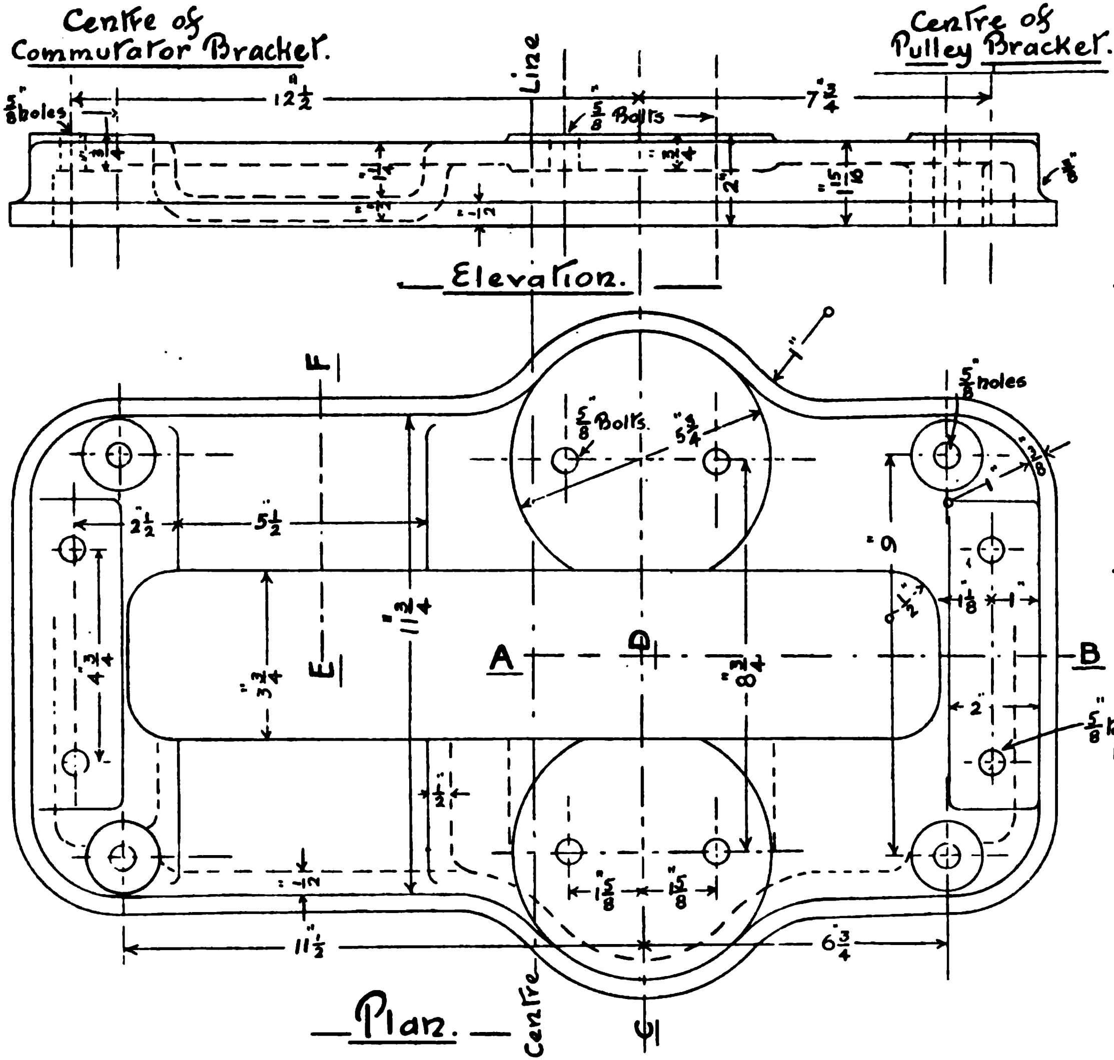
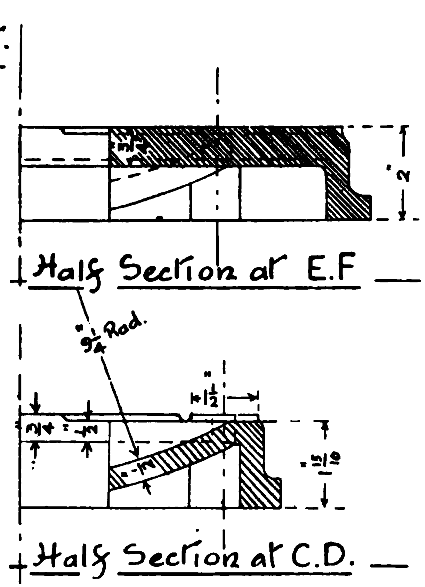


FIG. 727.

BED PLATE FOR DYNAMO (Drawing Exercise).



FIGS. 728, 729.



FIGS. 730, 731.

Instructions.—Draw the whole of the parts shown on the diagram, assembling the same as when in use. Draw a vertical sectional elevation (taken through centre line of engine on plan), a complete plan and a vertical section (projected from sectional elevation) taken through axis of crank shaft in elevation. Dotted lines need not be shown. **Scale half full size.**

BRACKETS FOR DYNAMO.

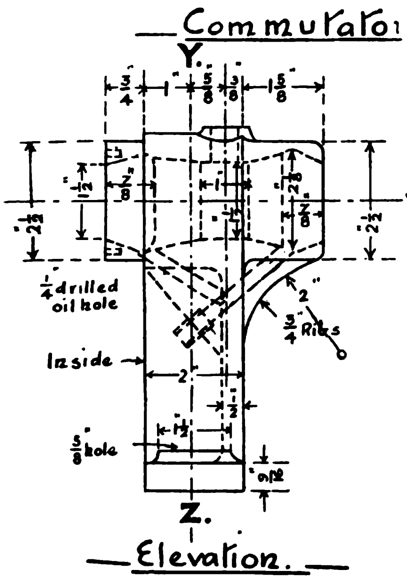


Fig. 732.

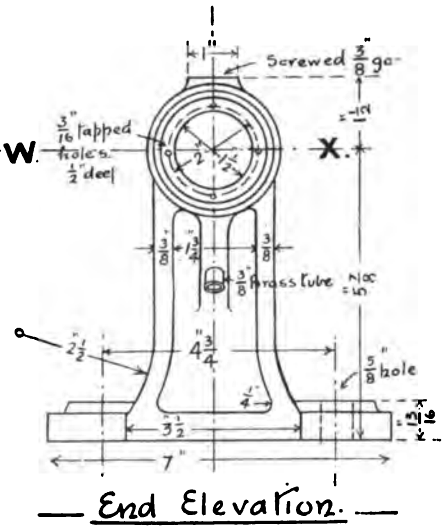


Fig. 733.

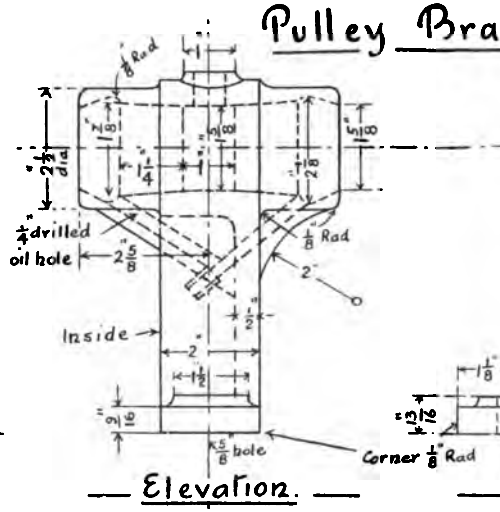


Fig. 734.

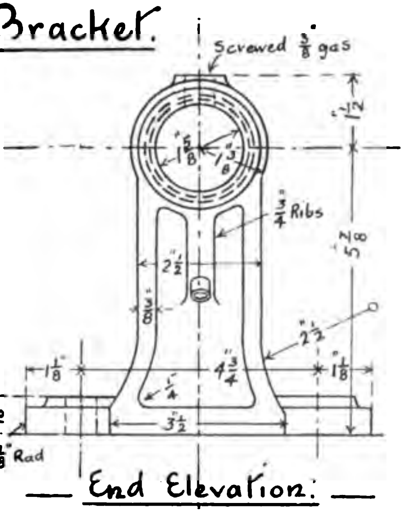


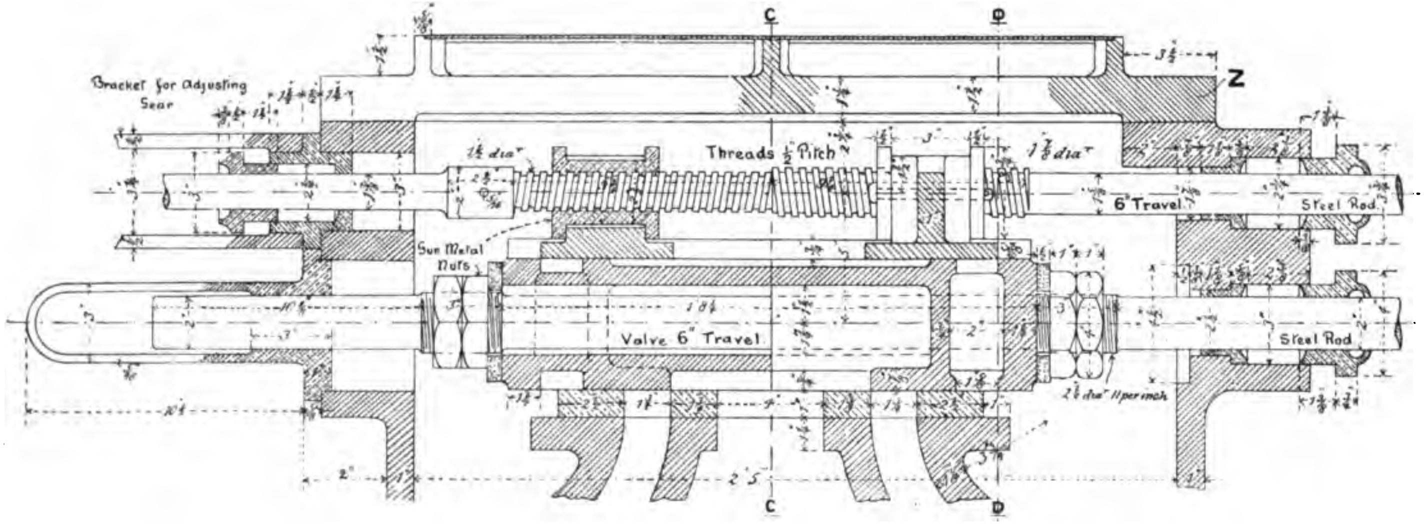
Fig. 735.

Advanced students should be able, with the assistance of the sketch showing the general arrangement in the upper right-hand corner of Fig. 738 and of the drawings of the petrol cylinders in the chapter on pistons, etc., which may be used as a guide, to make drawings of the complete engine.

266. Adjustable Loose Lathe Headstock.—This example was set at the C. G. Examination in Mechanical Engineering, Ordinary Grade, Part II., 1907, with the following:—

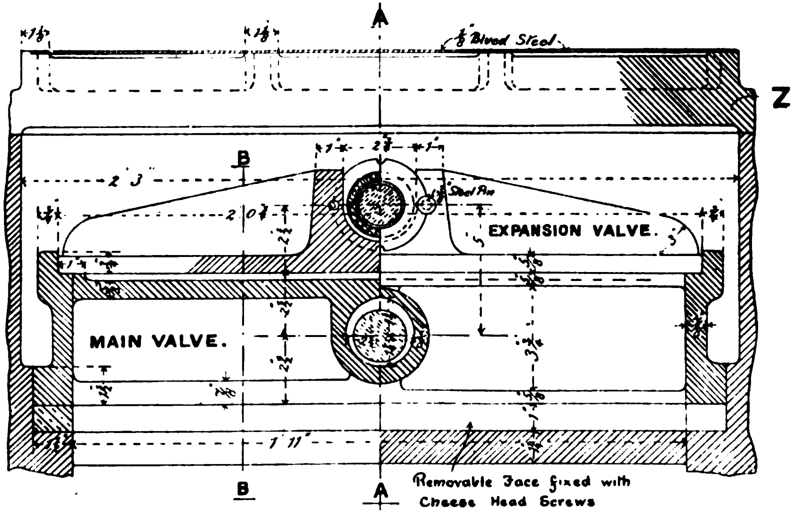
Instructions.—To obtain full marks for a drawing, it must be fully dimensioned, all views must be correctly projected, and the materials of which the parts are made indicated by sectional shading. *The figures are not drawn to scale. The drawings need not be inked in.* Candidates are recommended to draw on the blank side of the drawing paper.

EXPANSION SLIDE VALVE FOR COMPOUND VERTICAL ENGINE.



— SECTION ON LINE A A. — — SECTION ON LINE B B. —

FIG. 737



SECTION ON LINE CC. — — SECTION ON LINE DD. —

FIG. 736.

SOME DETAILS OF A 4 H.P. PETROL ENGINE.

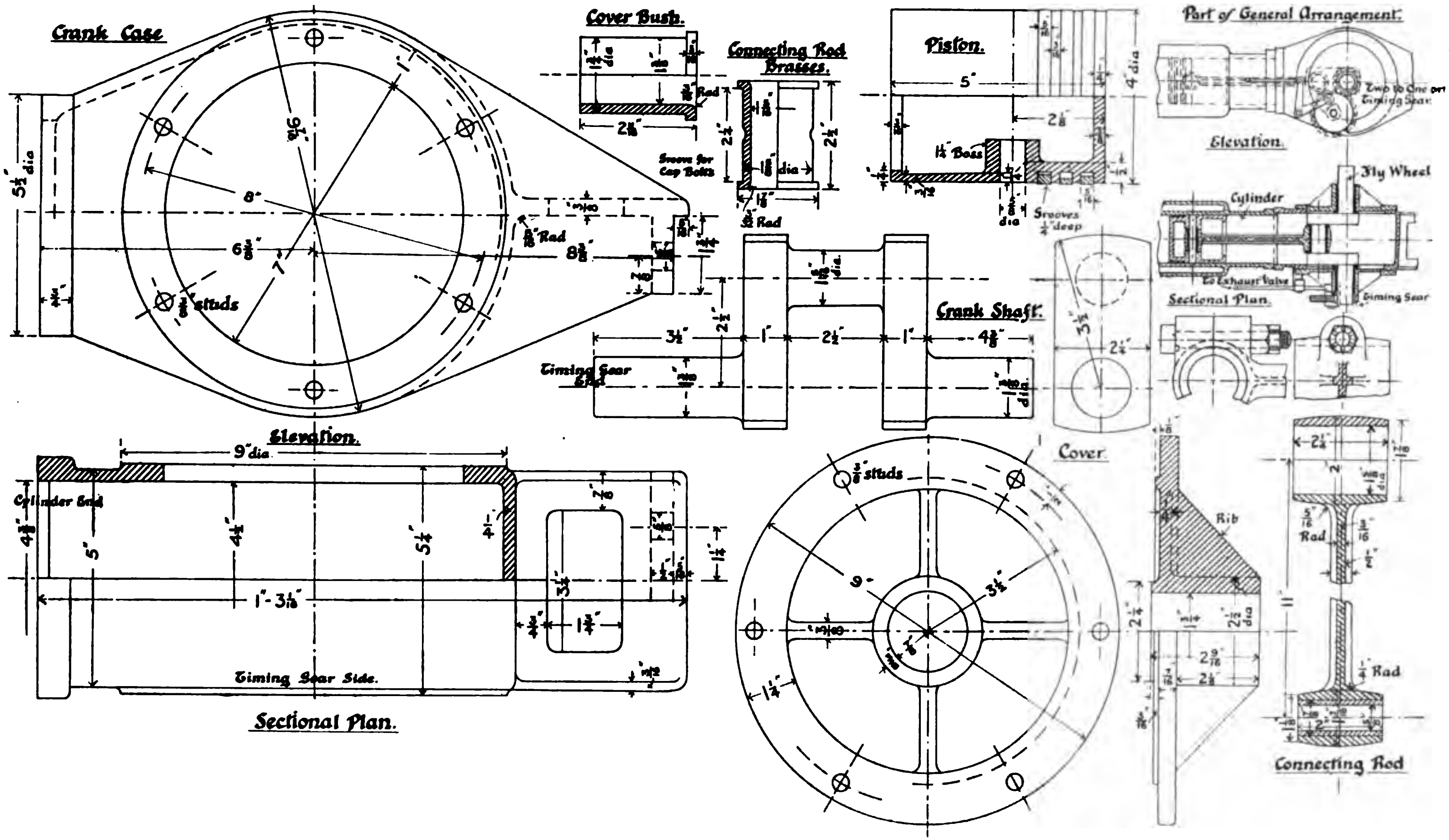
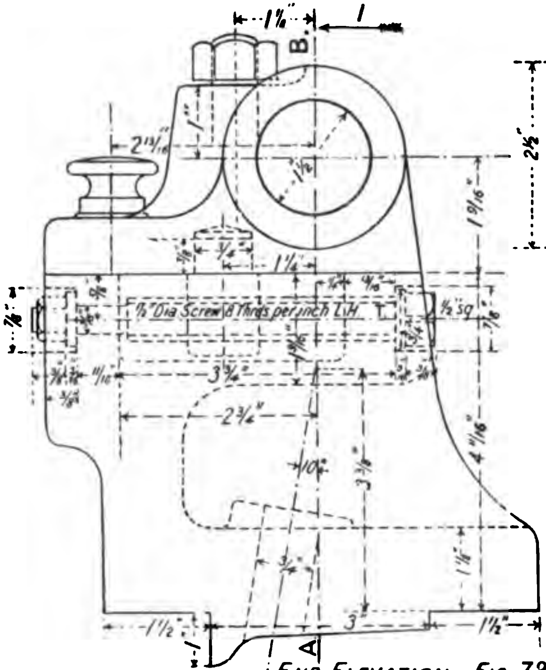
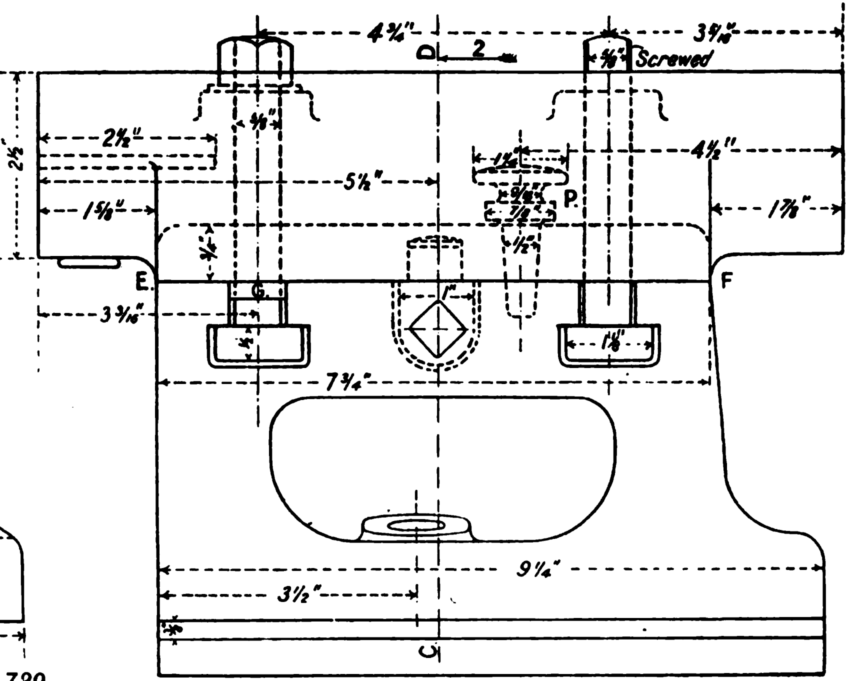


Fig. 738.

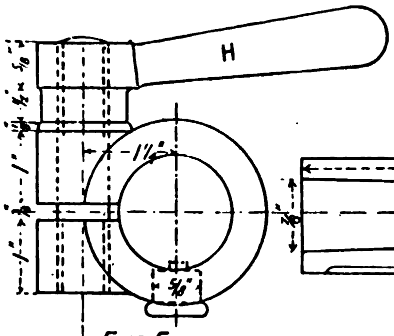
ADJUSTABLE LOOSE LATHE HEADSTOCK.



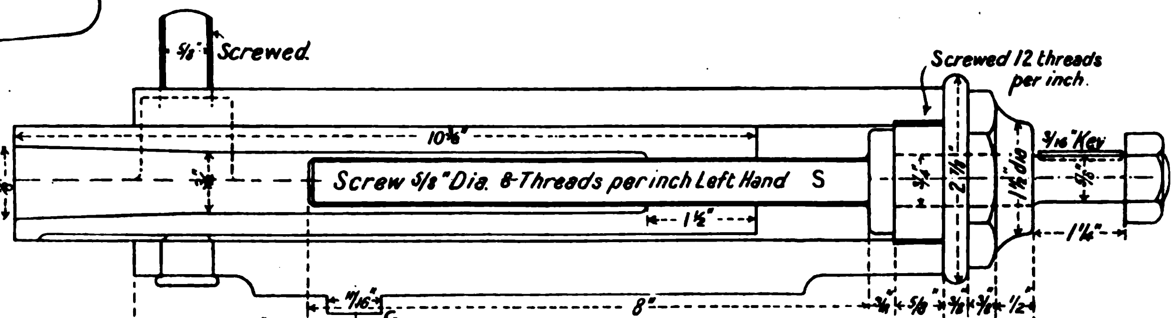
End Elevation Fig 739



Side Elevation Fig 740



End Elevation of Barrel Fig 741



Section through Barrel on A B Fig 742

Figs. 739 to 742 show four views of an adjustable loose headstock for a lathe. Draw to a scale of $\frac{3}{4}$ full size :—

(1) A section on the plane AB, looking in the direction of the arrow 1.
(2) A section on the plane CD, looking in the direction of the arrow 2.
(3) An end elevation of the "centre" end.
(4) A complete plan.

Add omitted detail.

267. Slide Rest for 9¼″ Lathe.—Advanced students will experience little difficulty in making working drawings from the dimensioned pictorial sketch of the unfinished slide rest shown in Fig. 743.

Instructions.—Draw two elevations and a plan of the slide rest, showing in section as much of the plan as is necessary to clearly display any important detail. You are to arrange a tool holder on the top slider, and show any other part that may be omitted. **Scale half size.**

268. Cylinder, with Meyer Expansion Valve, for 20 Horse-power Horizontal Steam Engine.—Eight fully dimensioned views of the cylinder and its fittings are shown in Figs. 744 to 751.

Instructions.—Set out the views shown to a **scale of 3″ to the foot, and make working drawings of the piston to a scale of half size,** and of the **main and expansion valves, bushes, etc.,** to a **scale of full size.**

SLIDE REST FOR 9¼″ LATHE.

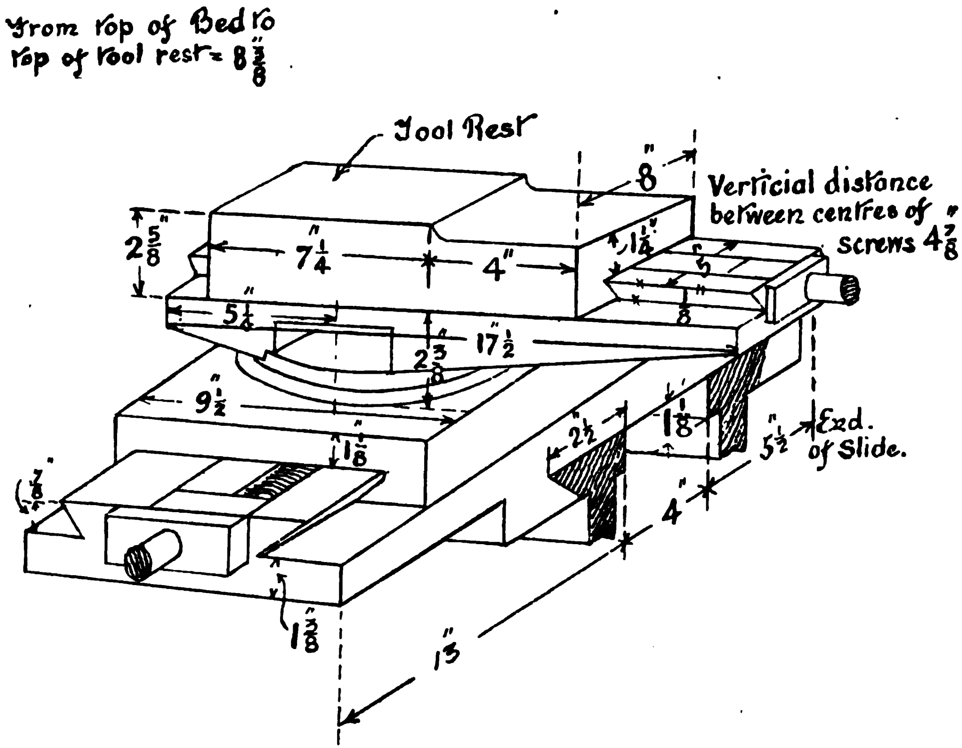
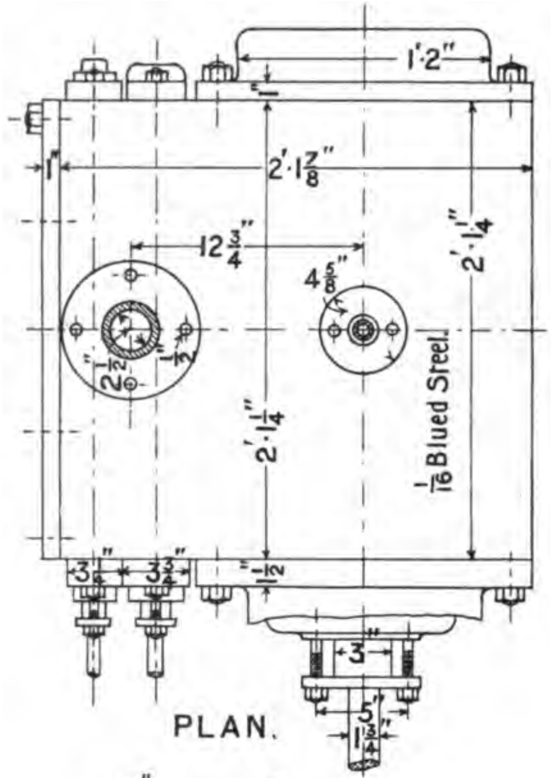
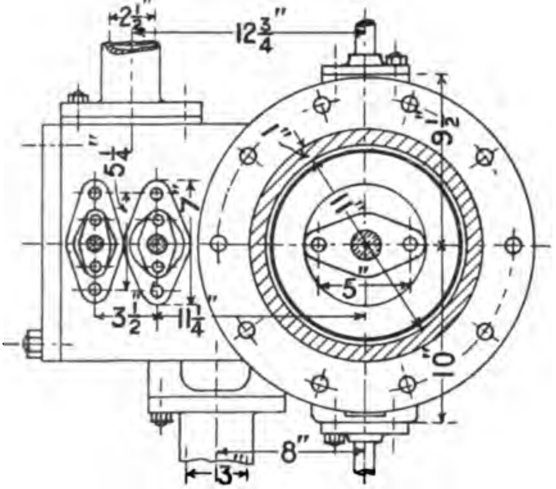


Fig. 743.

CYLINDER FOR 20 H.P. HORIZONTAL STEAM ENGINE.

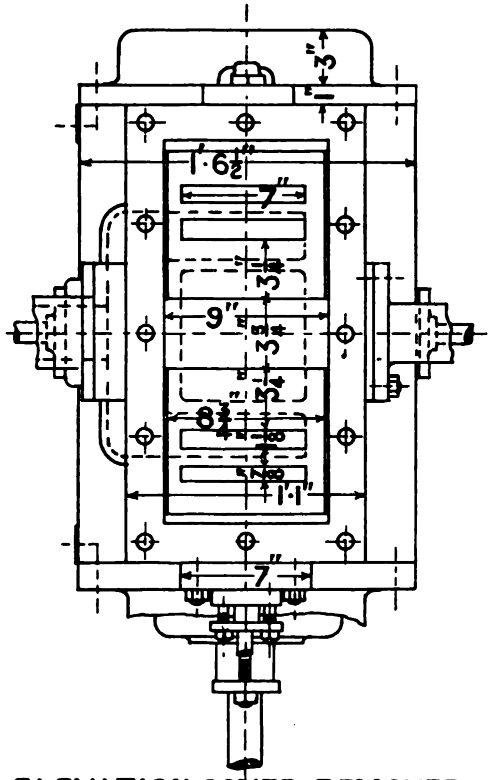


PLAN.

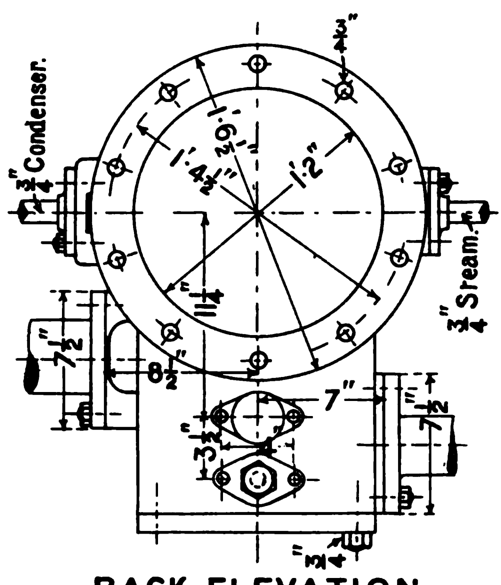


FRONT ELEVATION.

Figs. 744, 745.



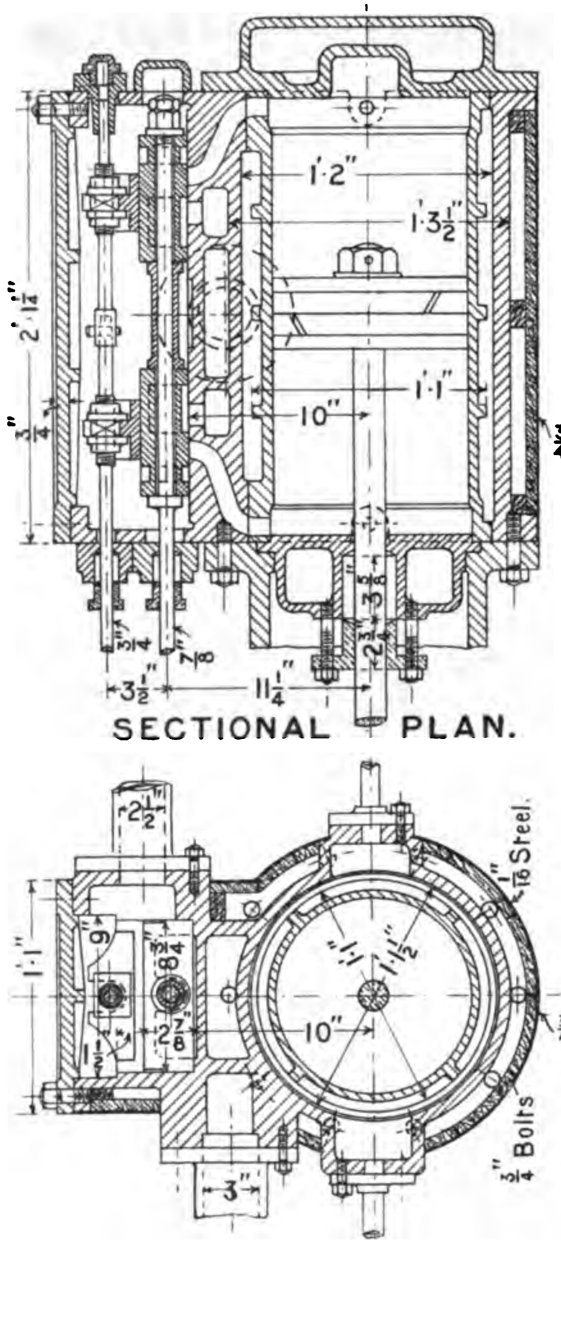
ELEVATION, COVER REMOVED.



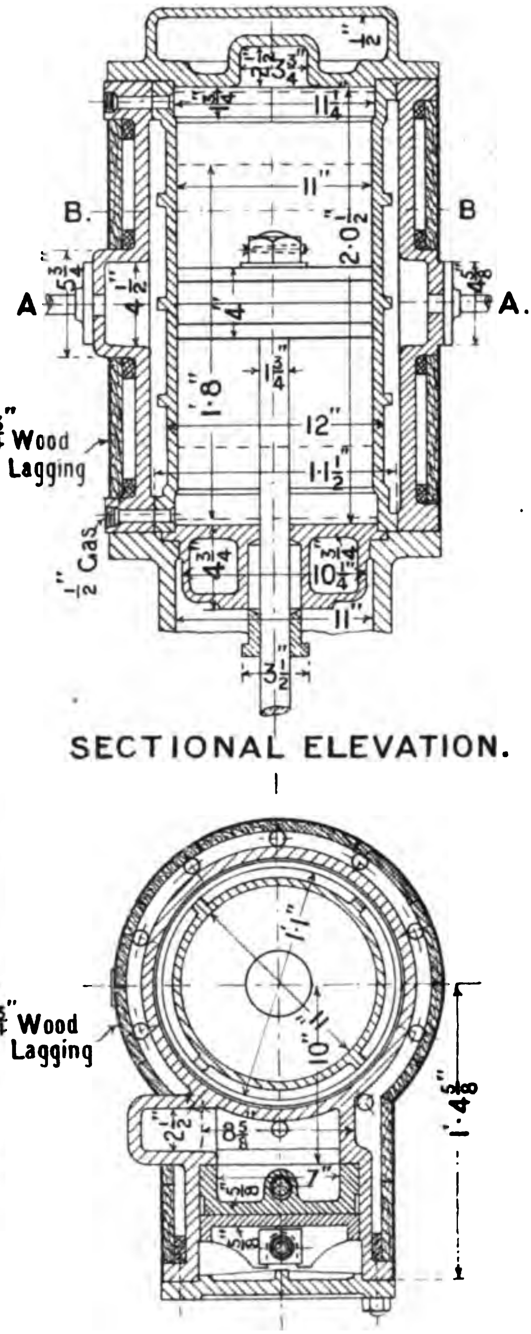
BACK ELEVATION.

Figs. 746, 747.

CYLINDER FOR 20 H.P. HORIZONTAL STEAM ENGINE.



FIGS. 748, 749.



FIGS. 750, 751.

CHAPTER XXV

PRINCIPAL MATERIALS USED IN THE CONSTRUCTION OF MACHINES

269. Cast Iron.—The crude metal derived from smelting common ores of iron with fuel in a blast furnace is *cast iron.* A strong blast of air acting on the burning fuel generates an intense heat, which gradually melts the iron. As iron ores are generally found mixed with earthy materials, which make them refractory, *fluxes* have to be used with the fuel to combine with the earthy materials and facilitate their fusion. When the ore is **calcareous** the flux employed has to be of an **argillaceous nature**, that is to say, to contain clay; on the other hand, if the ore contains **clay** the flux must be of a **calcareous** nature. This being so, it is occasionally possible to mix the two kinds of ore in proper proportions to enable the one to act as a flux to the other. When a flux is used it is tipped into the furnace with the fuel and ore, and it unites at a high temperature with the earthy matter of the ore, forming **slag,** setting the greater part of the iron free, which, as it fuses, falls by gravitation to the bottom of the furnace, and when a suitable quantity has accumulated it is allowed to flow out of a **tap-hole** on to a sand bed along a large groove in it (called a **sow),** from which at right angles it enters smaller grooves, or hollows, which form the moulds for the **pigs.** And these castings are commercially known as **pig iron.** During the smelting process the liquid iron absorbs and combines with a considerable quantity of carbon from the fuel, and is more or less contaminated with the impurities of the ore fuel and flux, hence the presence in cast iron of **carbon, sulphur, silicon, phosphorus,** and **manganese.** A portion of the carbon is **chemically combined** with the iron, while the remainder exists in the iron in the form of **graphite,** but the presence of carbon in the iron, whether in combination with it or not, determines its behaviour, giving to it its **fusibility,** which enables it to be remelted again and again for foundry purposes, and rendering the iron more liquid in the fluid state, and tougher and softer when in the solid state; the degree of fusibility depending upon the percentage of carbon which it contains, but an excess of carbon weakens the iron, and therefore the skill, experience, and judgment of the founder have to be exercised to secure by a suitable mixture of different sorts and qualities of iron the requisite degree of strength, softness, hardness, toughness, and closeness of grain for various kinds of castings. Strangely enough, the best results for both strength and elasticity are obtained by **mixing** a number of carefully selected different kinds of iron in the cupola; this gives higher tensile strengths than the average of the different samples when cast separately. When practically all the carbon is *combined* with the iron, the fracture of a freshly broken piece will have a silvery white colour, and the cast iron is **white,** and is found to be very **brittle and hard.** When only a little carbon is **combined,** and most of its particles crystallize separately, a fracture is **grey** in colour, and the iron is **weaker and more fusible.** **Silicon** apparently influences the form the carbon takes in cast iron, also the rate of cooling. The more slowly a casting cools the more graphite forms and the softer the iron. Usually, for commercial purposes, cast irons are divided into **seven varieties.** The greyer cast irons, containing the most graphite or free carbon, used for **foundry purposes,** are classed as Nos. 1, 2, and 3. The whiter and harder cast irons, Nos. 4, 5, 6, and 7, are used only for conversion

into **wrought iron**, No. 4 being occasionally used to close the grain and harden the metal of foundry mixtures. As the greyest iron, No. 1, is wanting in strength, most castings are composed of mixtures of Nos. 1, 2, and 3 in varying proportions, according to the judgment of the founder. No. 3, Scotch iron, is most generally used for **engine castings**, as it can be depended on for closeness of grain and strength, and it runs sufficiently fluid to make any casting. But generally, the stronger and larger the casting the smaller the proportion of No. 1 used. On the other hand, a larger proportion of No. 1 gives greater **fluidity** and causes the metal to run very thin and expand at the moment it solidifies, so that intricate forms and sharp corners of the mould are filled better. Nos. 5 and 6, called **forge irons**, often present a mottled appearance, as if a grey iron and a white iron had been melted and imperfectly mixed, hence it is often called **mottled pig**. No. 7, called **white forge**, is very hard, and silvery white in appearance.

270. Chemical Composition of Cast Iron.—The proportion of **carbon** in cast iron varies in different varieties from about 3 to about 4·6 per cent.,[1] and its effect, etc., is explained in the previous article. Mr. Bloxam, the famous authority on cast iron, gives the following as the composition of the kinds of iron in the foundry to which reference has been made:—

TABLE 12.—Chemical Composition of Various Kinds of Foundry Cast Iron.

	Bloxham.			Swedish Lily.	Foundry Glengarnock.	Spiegels.	
	No. 1 (Grey).	No. 2.	No. 3.			Ebbw Vale.	Ferro-Manganese.
Iron	90·24	89·31	89·86				
Combined carbon	1·02	1·79	2·46 }	4·603	3·677	3·734	6·588
Graphite	2·64	1·11	0·87 }				
Silicon	3·06	2·17	1·12	0·070	2·400	0·215	0·187
Sulphur	1·14	1·48	2·52	0·006	0·602	0·064	0·081
Phosphorus	0·39	1·17	0·91	0·015	1·010	0·088	0·059
Manganese	0·38	1·60	2·72	1·276	1·777	8·958	65·150

Whilst the ore is being smelted in the blast furnace a glassy **slag** is formed by the alumina, silica, and lime in the ore and flux; by aid of the heat, this floats on the molten metal and is run off near the bottom of the furnace. Part of the carbon of the fuel combines with the iron, the remainder combining with the oxygen in the air and ore, forming carbonic oxide and carbon dioxide, which pass out of the furnace at the top.

271. Strength of Cast Iron.—Probably no metal used by the engineer varies so much in strength and soundness as cast iron, and, as this material is so largely used in the construction of the machines and structures he is responsible for, no efforts on his part to get a sound knowledge of its physical properties should be spared. Cast iron of an inferior quality may have an ultimate tensile strength of 5 tons per sq. inch, or even less, but such exceptionally poor qualities have no value where strength is required; they

[1] For the methods employed in estimating the proportion of carbon in iron, see an article in *Technics*, April, 1904, by H. C. H. Carpenter, B.A., PH.D., on "The Estimation of Carbon in Iron-carbon Alloys."

may be used for balance weights, foundation blocks, or for purposes where weight alone is of consequence. On the other hand, a tenacity of 14 or 15 tons per sq. inch is sometimes reached, and in very exceptional cases 17 or 18 tons, and even 19·6 has been reached,[1] but the **average ultimate tensile strength of cast iron is about 7 tons per sq. inch.** Further, as the limit of its elasticity, as determined by short specimens of ordinary quality, is found to be only about one-third of its ultimate strength, it is *not considered* **safe to stress** *ordinary cast iron in tension to more than* **2 tons per sq. inch.** Even with the higher qualities, some authorities believe, and with sound reason, that the stress should never exceed 3 tons per sq. inch for a *statical* load.

As to **compressive strength,** it appears from Mr. Hodgkinson's experiments, that it varies from about 25 to 52·5 tons per sq. inch; averaging 38·5. And that the **average ratio** *of tensile to compressive* strength is **1 to 5·641.** More recent experiments give somewhat higher values.

272. Wrought Iron.—Wrought or malleable iron, which is very nearly pure iron, is obtained from cast iron by eliminating the greater part of the carbon in a **reverberatory furnace.** The pig iron sometimes undergoes two processes—one called **refining**, the other **puddling.** These are chemically the same; briefly, the former is usually done in a hearth termed a **refinery.** The pig iron and scrap are placed in alternate layers with coke upon a bed of ignited fuel at the bottom of the hearth. A blast is supplied at a pressure of about 1½ to 2½ lbs. per sq. inch, according to the quality of the coke. The charge is melted in about 2 to 2½ hours, and in about another hour the blast has sufficiently **oxidised the impurities** in the iron (when basic iron slags and hammer-scale are added the refining is accelerated) and a plate about 3″ in thickness is formed; the refined metal, being very brittle, is easily broken into pieces suitable for puddling. The fracture is a silvery-white; the top part being dull and cellular, and the lower part compact. This iron is ready for **puddling.** About 4 cwt. of it forms a charge for the reverberatory furnace, and in about half an hour it is partially melted, forming a pasty mass, which is stirred with iron tools so as to bring all parts under the oxidizing influence of the air and **fetling.**[2] The iron becoming less and less fusible as the carbon and impurities are removed by oxidation, the C forming, with the O of the air, CO_2, requires the temperature to be gradually raised.

The metal, which is now comparatively pure, is collected in balls or **blooms** weighing about 80 lbs., and, being now in a soft spongy condition, it is subjected to a hammering or squeezing, called **shingling**, and whilst these shingled blooms are hot enough they are rolled into rough **puddled bars,** which are of very inferior quality, having a tensile strength of about 9 tons per sq. inch only; they are not used by engineers, as they require to be further improved. This is done by cutting up the bars into short lengths, which are piled crossways into a **faggot** or *pile,* reheated, and hammered or rolled again, usually into bars which are commercially known as **merchant's bars.** This iron is still of low tenacity, and not very uniform in quality or structure; it is used for common girder work, gratings, ladders, fire bars, bearers, etc. The process of cutting, piling, reheating, and rerolling, may be repeated several times to give the iron a *fibrous structure,* according to the quality of the iron required. Thus **best bar** is made from faggots of merchant bars. Its strength is now much improved, and it is more uniform in quality; in fact, it is suitable for general smithing purposes, having a strength, if of good material, of 23 or 24 tons per sq. inch. The **best best**, or **double best**, is made from faggots of selected *best* iron, and **best best best,** or **treble best**, from faggots of *best best* iron. (Bars and plates of these qualities are commonly marked B; BB; and BBB respectively.) It has a very silky uniform fibre, and good qualities have a tensile strength of 25 to 27 tons, an elongation of 25 per cent., and a contraction at fracture of about 50 per cent., and it will bend double cold.

[1] Refer to Table 13.

[2] The action is assisted by the covering or fetling of the bottom of the furnace, formed of scales of oxide of iron.

Slabs from faggots of **selected scrap iron** are used to make up **heavy forgings**, new iron being seldom now used for this purpose. Among the best known qualities of wrought iron we have Swedish iron, and **Yorkshire iron** from Lowmoor, Farnley, Leeds Forge, and Bowling, used for the most difficult forgings for boiler plates which require flanging, for furnace plates exposed to furnace flames, etc.; and *treble best* **Staffordshire** is largely used for chains, boiler plates and general forgings, domes, and such parts of furnaces and chambers as are not exposed to the direct action of the flame; although slightly inferior to best Yorkshire, it is recognized as being of high quality. Whilst in **charcoal** iron we have the purest, and therefore the most soft and ductile quality.

273. Cold shortness in wrought iron, or *brittleness when cold*, is produced by the presence of a small proportion of **phosphorus**; and **red shortness,** *or brittleness when hot*, by the presence of **sulphur.**

Although **wrought iron** has in recent years been largely supplanted by mild steel for many purposes, it is still extensively used, particularly on account of its **weldable property**, for although it cannot be cast in moulds, it assumes, when heated, a sticky or viscous condition, so that when two or more pieces are brought together at the proper temperature they may be united by blows of a hammer or by pressure, or, in other words, *welded.*

274. Strength of Wrought Iron.—It is found that when iron bars or plates are rolled, the molecules of the iron are elongated or spread into a fibrous condition; this gives the metal (more especially when thin, as in boiler plates) a tensile strength in the direction of the fibres somewhat greater (about 7½ to 15 per cent.) than in a direction at right angles to them.[1] And the elongation is greater in the former than in the latter. The ultimate tensile strength of wrought iron ranges from about 18 to 27 or 28 tons per sq. inch; those qualities with the **greater strengths** tending to be **hard and steely.** Indeed, strengths of 32 tons have been produced; but such iron is **hard, brittle, and almost unweldable.** The contraction of the area of the transverse section where rupture occurs is usually taken as a measure of **toughness or ductility** of the metal. This contraction ranges from about 7 to 45 per cent. of the area. As a rule **bars** are **stronger than plates**, angles, tees, and like sections. Wrought iron in tension elongates about $\frac{1}{10000}$ of its length for each ton per sq. inch of its section, up to the *limit of elasticity*. The **elastic strength** (the strength up to its limit of elasticity) is not often less than **50 per cent. of the ultimate strength**, and may be taken at about 12 tons per sq. inch, in both tension and compression. The percentage of **elongation** (in terms of the length) before rupture occurs, is also important. Obviously, it will be greater for **short specimens** than for long ones, as most stretch occurs near the fracture, so it is necessary to state the length of the specimen in giving the elongation. Usually 8″ is the length for tensile tests. Ductile iron, such as is required for flanged plates or difficult forgings, usually has an elongation of 15 to 20 per cent., and a tensile strength with the fibre of about 25 tons.

Wire Drawing and Cold Rolling considerably increase the tenacity and hardness of wrought iron, but after annealing, it practically returns to its original strength and softness.

275. Steel. Preliminary Remarks.—We have seen that *wrought iron* contains very little carbon, an amount not exceeding some 0·4 per cent.; and that *cast iron* is rich in carbon, and may contain from about 3 to nearly 5 per cent. On the other hand, steel lies intermediate between cast iron and wrought iron, being pure iron combined with carbon and other elements, such as manganese, silicon, phosphorus, etc., each of which influences the physical properties of the metal, and some special qualities are alloyed with

[1] Fairbairn was apparently the first to discover by actual tests this difference; but, strangely enough, in his communication to the Royal Society, credited the strength across the fibres with the higher value, doubtless owing to some mistake in marking the plates.

such elements as nickel, chromium, vanadium, etc., to give them certain required properties. The hardest steels contain about 1·2 to 1·6 per cent. of carbon, and the mildest from about 0·16 to 0·4 per cent. The latter, called **low carbon steel**, much **resembles wrought iron**, which it has for many purposes supplanted, as we have before remarked. It is **weldable** and does not harden when suddenly cooled. With a little carbon, the steel is stronger and harder, and is used for rails, wheel tyres, etc.; but when the percentage of **carbon reaches 0·5**, the steel has the remarkable property of **hardening**. Steel is obtained either by the abstraction of carbon from cast iron, or by the addition of carbon to wrought iron, as we shall see. The former represents the cheaper qualities, and the latter the more expensive ones. We will now give some attention to the various kinds of steel in use, commencing with the former.

276. Bessemer Steel is made from grey pig iron, free from phosphorus and comparatively free from sulphur, containing a small quantity of manganese and silicon and a large proportion of free carbon. In the Bessemer process, there are essentially two operations, the conversion of molten cast iron into pure iron, and by the addition of a small and definite quantity of carbon, the turning of pure iron into steel. The pig iron is melted in a cupola, and run into a **converter** lined with firebrick, and mounted in hollow trunnions, through which air is blown through the metal for about twenty minutes, removing all the carbon: the oxygen of the air combining with the carbon of the iron forming CO_2, and in so doing burning out the carbon. From 5 to 10 per cent. of **spiegeleisen**,[1] an iron containing a known proportion of carbon and manganese, is then added, and the blowing resumed long enough to incorporate the mixture. The steel is then run into a **ladle**, and from the ladle to iron **moulds** to form **ingots**. These being more or less porous, are reheated,[2] and run through the five grooves of a clogging mill, or worked under the steam hammer, and finally rolled or forged into the required shape. This steel, which is named after Bessemer, the inventor of the process,[3] is largely used for structural purposes, tyres, rails, etc.

Fairly good steel can be made from iron containing **phosphorus**, by the **Thomas-Gilchrist** process. The phosphorus is absorbed by the converter lining, which is prepared from magnesium limestone, the product is known as **basic steel**, the original metal being called **acid steel**.

277. Siemens-Martin or Open Hearth Steel.—This steel is produced by melting (heated by gas to an intense violet heat) a certain quantity of pig iron in the hearth of a *Siemens reverberatory furnace*, and **adding wrought iron** till the bath attains the desired degree of carbonization, or by mixing cast iron and certain kinds of iron ores. The oxides and free oxygen are removed, and carbon and manganese added by the introduction of a small quantity of **ferro-manganese** (a somewhat similar substance to spiegeleisen), which is rich in carbon and manganese. The amount of carbon left in the metal is ascertained by testing a small quantity, which is removed by a ladle, quenched, broken up and tested by the chemist on the works. If found satisfactory, the charge is tapped and the metal run into ingot moulds. The **operation** is **slower** but **more completely under control** than that of Bessemer's. The regularity and ease with which any grade of steel required can be produced by it has led to its being freely adopted; in fact, it is now the most general method of producing on a large scale steel of good and uniform quality at such a comparatively low cost that it can compete in price with Bessemer steel. This is largely due to the modern practice of using a basic lining for the Siemens furnace for the

[1] White pig iron containing 5 to 10 per cent. of manganese is known as Spiegeleisen. This gives to the metal the small proportion of carbon required. and the still smaller quantity of manganese which seems to be so essential for the production of good steel.

[2] The ingots are usually taken from the moulds when their skin has solidified (their interior being still more or less liquid), and placed in an oven to soak and allow the temperature to become more equalized. They then have a bright cherry-red heat.

[3] See *Engineering*, Nov. 22, 1901.

dephosphorization of pig iron. A grade of this steel with a tensile strength of 26 to 32 tons per sq. inch, and not less than 20 per cent. elongation in 8″, is largely used for the crank and tunnel shafts of merchant ships and war vessels, etc.; and steel plates, bars, and forgings are made almost exclusively from ingots run from Siemens furnaces or from Bessemer converters.

278. Mild Steel.—Usually in speaking of *mild steel* we refer to such steels as are worked up in bars, plates, angles, etc., from **Siemens open hearth or Bessemer ingots.** The ingots are reheated and hammered into slabs, which are again reheated and rolled into plates or bars. Such steels *do not harden* perceptibly when heated and quenched in cold water. Owing to the low percentage of carbon they contain (0·15 to 0·5 per cent.) they resemble wrought iron, and can be easily welded,[1] with the additional advantage that plates of much greater area and weight, and bars of much greater length [2] can be obtained without extra cost per unit of weight. Mild steel has now superseded wrought iron for many purposes, particularly for boiler plates and stays, bolts and shafting, engine details, etc. The **lower the ultimate strength** of mild steel the **higher is its elasticity**, but it is a necessary condition that boiler plates must be **ductile.** The best Yorkshire iron plates stretch about 18 per cent. before rupture, and it is found that steel plates with about 20 per cent. of elongation have not a tensile strength greater than 30 tons per sq. inch. But this steel is largely used for boilers, as *it has nearly double the elasticity of ordinary iron boiler plates, and nearly* 50 *per cent. greater strength.* For parts of a boiler that are flanged a somewhat softer metal is used, generally one with an ultimate elongation of 25 per cent., and a tensile strength of 26 to 28 tons per sq. inch. Such engine parts as piston and connecting rods, shafts, valve rods, are often now made of mild steel forged from Siemens or Bessemer ingots. And ordinary mild steel bars having an ultimate tensile strength of 30 to 32 tons per sq. inch, and an elongation of at least 25 per cent., are used for boiler stays, studs, bolts, etc. A harder steel of 35 to 40 tons tensile strength, and 15 per cent. elongation, being used for pins and such like pieces.

279. Steel Castings, or *cast in steel*, means that the object is cast in form by mild melted steel being poured into a mould. When *bar blister steel is* **melted in crucibles** and poured from them into the mould, we get **crucible steel castings.** But large steel castings, such as beams for stationary engines, stern posts, propellers, large shafts, pistons, cross heads, standards for steam hammers, large stop valves, etc., etc., are now made by more direct methods and of less cost, the furnace being charged with pig iron, scrap steel, and broken ingot moulds.[3] Steel castings have the disadvantage of not being perfectly sound, owing to the pores or blow-holes in the metal, which may be below the surface and therefore out of sight, but more often they are on or near the surface, and can often be cut away in machining. In this respect they are also superior to iron castings, as blow-holes in the latter may be in the body of the casting. Apparently, any want of density in steel castings, owing to the presence of pores, can only be dealt with by subjecting the fluid metal to great pressure, on the Whitworth principle; but, of course, the cost of this treatment is prohibitive for most jobs. Notwithstanding this disadvantage, castings in steel are now produced without rolling, hammering, or other mechanical treatment, which are **superior in ductility** and **strength** to castings in any other metals in ordinary use, particularly cast iron, and we may safely look forward to further improvements in quality, and to a great extension in their use. In the opinion of some engineers, where hardness and resistance to wear are concerned, the castings made by English manufacturers are unsurpassed, but tougher and more ductile castings can be got from the Continent at an extra cost of about 50 per cent.

[1] In welding steel, care should be taken that the pieces to be united contain the same proportion of carbon or the welding temperature will be different.

[2] Steel bars are rolled up to 150′ in length, and plates with an area of 70 sq. feet and more.

[3] A full charge for an open hearth furnace may consist of hematite pig 8 tons 8 cwt., Weardale pig 1·5 tons, scrap steel 2·5 tons, broken ingot moulds 1 ton, broken skulls and scrap 2·5 tons, Elba ore 2·75 tons, manganese (80 per cent.) 2·21.

It would be impracticable or too expensive to forge certain parts of **motor-cars**,[1] such as *trailing wheels* of *heavy cars*, and *motor-buses*, *brake discs*, *chain wheels*, etc., but these can with advantage be steel castings. Such castings are, as a rule, made of a *medium* hard quality, the tensile strength of which is about 78,000 to 85,000 lbs. per sq. inch, and the elongation 18 per cent. for 2" length and 1" diameter.

280. Motor-Car Steels.—One of the most remarkable features of the development of the motor-car, is the wonderful improvements in the qualities of steel which have been made by steel makers, in meeting the demands of motor-car constructors for materials of the highest excellence, both as regards tensile and elastic strength. These qualities are usually measured by **static** and by **dynamic** tests to determine the **resistance to shock** and the **endurance of fatigue**, the great importance of which is now well understood. We have already given some particulars of high-class steels, but the new steels manufactured for motor-car work demand special attention. Probably no English firms have done more, if as much, in this movement than the famous one of Vickers Sons and Maxim, and, more recently, Messrs. Willans and Robinson, so, therefore, some particulars of the special steels they manufacture for this work should be referred to. Messrs. Vickers and Co. truly remark that: "In the construction of all modern machinery, the most suitable material for use is that which *will combine with a sufficiently high factor of safety the least possible weight* in any given part." For a steel to have high tensile strength is not enough. To be suitable and safe in machinery undergoing severe shocks, a steel must have *three main qualifications*. It must, in the first place, have a **high limit of elasticity**, so that it will *be able to endure high stresses without deformation*. It must also be **tough and ductible**, so that it can receive excessive and suddenly applied shocks without undergoing breakage. And, finally, it must be, as far as possible, **unaffected by long-continued vibration.** The introduction of new steels possessing these qualifications makes possible a very appreciable reduction of weight in many parts of a *motor-car*.

281. Copper.—The most important and useful metal used by the engineer next to iron is copper. Its ores are very widely distributed, being found in almost every part of the world. It is a metal which is both ductile and malleable when hot and cold, but as it possesses the latter quality in a higher degree than the former, it is used to greater advantage when rolled, hammered, or worked into sheets, cylindrical pipes,[2] hemispherical pans and such like forms, than when drawn through a drawplate into fine wire. It is possessed of considerable elasticity and *strength* when wrought, its tensile strength being about 15 tons per sq. inch. But in the ingot or cast condition, it contains much oxide and many cavities, therefore it is not so strong, often breaking easily with less than half the above tension. When pure it may be worked up by hammering or drawing to a state of great strength and toughness.

When hammered or **worked cold** copper becomes brittle, but it is restored to its proper degree of toughness by heating to about 500° F., or in other words, by annealing. When heated to redness it can be drawn down, upset and **forged**, but if overheated the surface, by exposure to the air, becomes converted into black scales of peroxide. Although copper loses strength as its temperature is increased, being at its best when cold, and is affected by the use of sulphurous coal, it is still used to some extent for locomotive furnace boxes. The ultimate strength of copper may be taken as follows:—

[1] An abstract of an article, "Locomotive Parts of Cast Steel," by Mons. du Bousquet, which appeared in the *Revue Générale des Chemins de fer*, is given in the *Proc. Inst. C.E.*, vol. clxviii. p. 375, in which the author, in referring to some experiments on the use of cast-steel piston heads, guide bars and their supports, coupling rods, and brake gear, etc., in actual practice, concludes that "the experiments prove that *cast-steel parts may be safely used in many cases for which forgings have always been considered necessary*."

[2] Copper pipes of 6" diameter and upward are made from sheets rolled or hammered into the required form and brazed at the seams, the joints being practically as strong as the original sheet. Smaller pipes are usually made by drawing, but these cannot be relied upon being of uniform thickness.

STRENGTH OF COPPER.

When carefully drawn into wire	38,000 to 60,000 lbs. per sq. inch.
Pure Wrought, Copper bolts	36,000 „ „
Ordinary „ „	33,000 „ „
Copper Castings	19,000 to 26,000 „ „

Copper when employed by itself is largely used for many purposes,[1] but when combined with other metals to form **alloys**, it is more extensively used for engineering purposes. **Boron** appears to affect copper much as carbon does iron; wire has been made of such alloy with a tensile strength of 27 tons per sq. inch, and without loss of electrical conductivity.

For some years now a process (**Elmore's**) has been at work by means *of which copper is deposited by electrolysis*, the metal being pure and remarkably strong. Bars of copper are melted in an ordinary furnace and granulated by being run into cold water, being afterwards placed on a copper tray at the bottom of a tank, which serves as the anode, or positive terminal; revolving on its horizontal axis above this tray is a copper cylinder, constituting the cathode, or negative terminal; a solution of sulphate of copper, or blue vitriol, is the electrolyte, and in this the revolving cylinder is completely immersed, contact being made with a copper brush. An agate burnisher, pressing upon the deposited surface, is automatically traversed from end to end, and it is claimed that this **burnishing action** gives to the metal its remarkable tensile properties.

282. Tin is seldom used alone by the engineer, as its tensile strength is too low (about 2·1 tons per sq. inch) and its cost too high, but as one of the chief constituents of gun-metal or bronze, it is of great value. Owing to its immunity from the corrosive action of salts and acids, it is used as a protective covering to other metals. The Admiralty, and some Mercantile Shipping Companies require all **condenser tubes** to be coated with tin, when fitted in **iron** condensers.

Thin sheet iron coated with tin, known as **sheet tin**, is used for oil feeders, lamps, and for liners or distance pieces between brasses.

283. Lead is to a small extent used as a constituent of certain alloys, as we shall see, but for many purposes it is used alone. Its ductility, and therefore the ease with which it can be bent to any form, and its resistance to the corrosive action of sea and bilge water, fit it for use as **bilge piping**, and for emptying and filling the ballast tanks of ships. It is also used for **jointing** pipes when their flanges are rough or uneven. Sheet lead is used for covering the **engine-room floors** of ships when they are made of wood, and to protect the covering of boilers from wet.

The **tensile strength** of lead piping is 1 ton per sq. inch, and that of sheet lead 0·8.

284. Zinc is largely used to alloy with copper to form brass and other alloys. It is also employed as a covering for iron to protect it from the action of the atmosphere or of sea water, etc.; being much cheaper than tin and easily applied to iron to **galvanize** it,[2] it is used on a much more extended scale than tin is.

It has long been known that a **galvanic couple** will prevent **corrosion in marine boilers** and hot wells, so blocks of zinc, or the residuum from the galvanizing bath (called *hard spelter*), are placed in metallic contact with the iron of the boiler in such places as experience proves requires protection. Of course the purer the zinc the more perfect the action.

[1] Refer to Arts. 136 and 145.

[2] The iron is cleaned by dilute acid and friction, it is then heated and plunged into a bath of melted zinc covered with sal-ammoniac, and is stirred about until the surfaces become alloyed with zinc. Mallet recommends an amalgam of zinc 2202, mercury 202, and about 1 of sodium or potassium; this melts at 680°. The cleansed iron is dipped in this, and removed as soon as it reaches the temperature of the alloy.

285. Gun-metal, or Bronze, is an alloy of copper and tin (and sometimes a small proportion of zinc) in varying proportions; there being no particular mixture to which this name properly belongs. Strangely enough, when copper and tin are alloyed in good proportions *harder metal than either* of them is produced, with a *density greater than the mean density of the constituent.* The metal is also more fusible and less likely to corrode than copper. It is found that with castings rapidly cooled (**chilled**), the density, strength, and toughness are increased, due to the composition becoming more uniform. From experiments made at Woolwich,[1] upon alloys of the usual proportions, the following results were obtained :—

TENSILE STRENGTH OF GUN-METAL.

12 parts	of copper	and 1	of tin	29,000	lbs.	per sq. inch.
11	,,	,, 1	,,	30,700	,,	,,
10	,,	,, 1	,,	33,000	,,	,,
9	,,	,, 1	,,	38,000	,,	,,

The last of these compositions is the best known; it is fairly hard and very tough.

Although much higher values have occasionally been registered for special mixtures,[2] 33,000 lbs. or between 14 and 15 tons per sq. inch, may be taken as a general average for good gun-metal. Compared with steel and iron, gun-metal offers a small *resistance* to compression. This resistance is found to vary very much with the perfection of the alloy, the rate of cooling employed to prevent the separation of the tin, and the amount of fluid pressure in the mould due to the height of the *deadhead.* The **elastic limit in compression** is about 14,000 and the **ultimate strength** 27,000 lbs. per sq. inch.

The general effect of **tin** in the alloy is to **increase its hardness.** It also whitens the colours. **Zinc** alloyed with copper in small quantities increases fusibility without reducing the hardness. In larger quantities it prevents forging when hot, but increases malleability when cold. Although a small quantity of zinc added to common *bronze* makes it mix better, it is seldom used in *gun-metal.*

For **heavy bearings** hardness is considered to be of more importance than strength, although of course a good strength is required. A suitable metal is formed of 79 per cent. copper, and 21 of tin; its tensile strength is nearly 14 tons per sq. inch. The alloy specified for propellers and all bronze castings by the Admiralty (known as **Admiralty bronze**) is 87 per cent. copper, 8 per cent. tin, and 5 per cent. zinc, giving a tensile strength of over 14 tons per sq. inch.

Copper and tin mix well in almost all proportions. The alloys, or proportions, given in the above Table are among the best known ones.

286. Phosphor Bronze is an alloy of copper and tin to which some lead and phosphorus have been added; it is harder than ordinary gun-metal, of superior strength and very close-grained, the usual proportions of the above being 79, 10, 10, and 1, respectively. But its strength, hardness, and ductility can be varied by altering the proportions. Its strength, etc., appears to be as follows: **soft quality** elastic limit equals about 5 tons, and ultimate strength 22 tons per sq. inch; with about 30 per cent. of elongation. **Hard quality** elastic limit equals about 25, and ultimate strength 33 tons per sq. inch, with of 3 or 4 per cent. of elongation.

It bears remelting better than gun-metal, but depreciates after many repeated re-meltings. It is very red-short, and liable to

[1] Anderson's "Strength of Materials," p. 85.

[2] Dr. Thurston found that the alloy which gave the maximum strength of 70,000 lbs. per sq. inch was one which consisted of copper 55, zinc 43, and tin 2 per cent., but this could hardly be called gun-metal.

crack. When drawn into **wire** it has the **extraordinary strength** of 100 to 150 tons per sq. inch *unannealed*. But when annealed its strength is reduced by about 50 per cent. As it is a good metal for resisting shocks it is used with advantage for bearings for rolling mills, railway axles and crank shafts (particularly *motor-car* ones), and for such pieces as propeller blades and pump rods. The strength of this metal is also somewhat less affected by heat than gun-metal is, and it can be rolled into extremely thin sheets, and is then useful for the valves of air pumps, etc.

The following table gives particulars of the ultimate and elastic strength, etc., of the principal materials used by the engineer:—

TABLE 13.—ULTIMATE AND ELASTIC STRENGTH OF MATERIALS.

Material.	Ultimate tensile strength.	Ultimate compressive strength.	Elongation per cent. on 8″ length.	Elastic limit.	Young's modulus of elasticity (E).
	Tons per sq. inch.	Tons per sq. inch.		Tons per sq. inch.	Tons per sq. inch.
Aluminium castings, about 98 per cent. pure	5 to 7	—	2 to 3	—	—
Common grey cast iron	7 to 9	44 to 47	—	—	—
Special cast iron for cylinders, etc.	10·5 to 13·5	47 to 50	—	—	—
Malleable castings	16	—	—	—	—
Wrought-iron plate	22 to 23	16 to 18	—	—	—
Good welding iron, small forgings	22 to 24	16 to 18	14 to 18	12 to 16	12,700
Siemens-Martin, mild forged steel	24 to 27	19 to 21	20 to 25	12 to 19	13,600
Siemens-Martin, forged steel for shafts	29 to 35	23 to 28	20 to 25	12 to 22	14,000
Tool steel unhardened	48 to 57	—	—	35 to 40	14,000
Siemens-Martin, steel castings	25 to 32	20 to 28	18 to 20	12 to 19	13,600
Boiler plates of mild steel	24 to 28	19 to 24	20 to 25	16	13,600
Best gun-metal, bronze for valves, etc.	12 to 19	—	10 to 20	—	5,700
Rolled brass	9·5	—	—	—	7,000
Muntz metal	22	—	—	—	—
Manganese bronze (bolts)	25 to 32	—	20 to 45	—	—
,, ,, (propeller blades)	19 to 29	—	15 to 25	—	—
Delta metal	22 to 24	—	11·5	—	6,350
Aluminium (cast)	8	—	—	—	—
,, (sheet)	12	—	—	—	—
Copper (bolts)	17	—	—	—	—
,, (plates)	13 to 15	—	38	9	7 000
Oak (with grain)	7	4·2	—	—	760
Teak (Indian)	6·7	5·4	—	—	1,070
Pine (with grain)	7	2·8	—	—	760
Elm (British)	6·2	—	—	—	—
Ash (with grain)	7·6	4·2	—	—	630
Hornbeam	6·7	5·4	—	—	—
Lignum vitæ	7·1	4	—	—	—

The strength in shear of most of the above metals varies from 0·7 to 0·9 of the strength in tension; 0·8 may be assumed without serious error.

CHAPTER XXVI

MISCELLANEOUS

TABLE 14

Greatest working pressures, p, per sq. inch of projected area[1] ($L \times d$) on various bearing surfaces (Unwin and other authorities).

	Pressure per sq. inch in lbs.
Crank pins of shearing machines, slow speed, intermittent load	3000
Cross head neck journals (intermittent load, oscillating motion), the higher pressures for locomotives and destroyers	800 to 2100
Gudgeon pins of petrol engines	800 to 1000
Crank pins, small land engines	150 to 200
„ „ marine engines	400 to 500
„ „ fast land engines	500 to 800
„ „ slow land engines	800 to 900
„ „ torpedo boats and destroyers	850 to 1000
„ „ locomotives	1200 to 1800
„ „ and crank shaft journal of petrol engines	350 to 400
Locomotive axle boxes—	
Passenger	190
Goods	200
Shunting	220
Locomotive tender and carriages	300 to 380
Main crank shaft bearings, according to speed, fast to slow, as follows:—	
„ „ „ ordinary freight steamers	200 to 225
„ „ „ quick running steamers	225 to 300
„ „ „ ironclads and large cruisers	250 to 350
„ „ „ small light cruisers	350 to 400
„ „ „ torpedo boats, steam tugs, etc.	400 to 550
Fly-wheel shaft bearings (unvarying load)	150 to 250
Eccentric sheaves, stationary engines	60
„ „ marine practice	70 to 140
Line shafting on gun-metal steps	200
„ „ cast-iron steps	50
Eccentric straps	70 to 140
Pivots, wrought-iron shaft on gun-metal step	200 to 700
„ cast-iron shaft on gun-metal step	200 to 450
„ wrought-iron shaft on lignum vitæ bearing	1000 to 1400
Collar thrust bearings for propeller shafts (according to speed)	50 to 80
Slides, cast iron on Babbit metal	200 to 300
„ cast iron on cast iron (according to speed, fast to slow)	40 to 100
Steel or iron shaft on lignum vitæ (water lubrication)	350
Faces of link blocks	220 to 350
Pins of „ „	550 to 1000

Thurston's rule relating to pressure and velocity is very important as a guide. It is, *the product of the rubbing speed in feet per minute and the pressure in lbs. per sq. inch, should not exceed* 50,000.

[1] Deducting area of oil grooves.

287.—Questions in Machine Construction and Drawing.—The following questions have been selected (and more or less modified) from the Polytechnic Annual Examination Papers set during the past few years by the author:—

QUESTIONS SUITABLE FOR EXAMINATIONS AND HOME WORK

Stage 1 (or First Year's work)

1. Show two methods of connecting two wrought iron plates at right angles to each other.

2. Make a dimensioned sketch showing the proportions of cast iron wheel teeth, and name the various parts and curves used. Pitch 3″.

3. Give sketches and a description of an adjustable pedestal with brasses. How would you lubricate it, and how adjust it when worn, say ⅛″ in a downward direction?

4. Sketch and describe the action of a syphon lubricator, and state the kind of bearings it is used on.

5. Illustrate two methods of connecting together the ends of two circular bars. One method to be suitable for an alternate push and pull, the other for a twisting action. The bars require to be easily disconnected.

6. Describe two forms of lubricator. Explain the action of each and on what kind of bearing you would use it.

7. Show three methods of locking nuts. Under what conditions is it desirable to use them?

8. Give sketches of the various forms of keys used. What are the proportions and tapers, and how are they used? Describe a "loose collar with set-screw," and say what it is used for.

9. Sketch two or three forms of cranks, and show how the pins are connected.

10. What is the object of the snug and the cotter, shown in the drawing example (Fig. 709), and why are they used?

11. How would you adjust the connecting rod end when the brasses had worn, say ¼ inch, in the direction of the length of the rod? What is the object of the circular projection at the back of each brass, and what is it called?

12. Show two methods of jointing two circular rods, one joint to be suitable for a pulling action and capable of adjustment, the other to be formed so that one of the rods may have a straight line motion and the other a motion through an angle of 20°.

13. Sketch and describe how you would make steam tight (*a*) the piston rod of a steam engine, pressure 80 lbs., (*b*) the ram of an hydraulic press water-tight, pressure 700 lbs. per sq. inch.

14. Sketch the different forms of key used, give the proportions of each, and explain what class of work each is specially adapted for. Also sketch a gib and cotter, and say of what material these details should be made.

15. The hole in a solid bearing has worn down ⅛″: how would you proceed to restore it to its original condition? What is a chipping piece, and a long hole? and where are they used? Give sketches.

16. Referring to the drawing exercise (Fig. 427).—How would you alter the height of the centre from 3¾″ to 3⅞″? and how would you adjust the brasses when worn down ⅛″? What is the bearing used for?

17. Sketch a ¾″ Whitworth bolt.
 „ a ⅞″ Snap head rivet.
 „ a Taper pin, split.
 „ a Snug. Say of what each would be made and where used, and figure each proportionally.

18. Referring to the drawing example (Fig. 682).—(*a*) What are the various portions of the cross head made of? (*b*) Why are the bolts turned smaller in the central portion than at the ends? (*c*) Describe the lock washer and how it is used. (*d*) What is the cross head used for?

19. Sketch, in plan and section, a double riveted butt joint, diagonally riveted with double cover plates. Figure it proportionally. What is chain riveting? Compare the two arrangements.

20. Sketch a half crank, showing how the pin is fitted and the crank fixed to the shaft. What should each part be made of?

21. Sketch two teeth of about 2″ pitch, suitable for ordinary cast iron wheels. Figure them proportionally, writing on names of parts and curves used.

22. Sketch (*a*) a gib head sunk key, give proportions and taper; (*b*) a ¾″ gland stud, say how many threads per inch it has, and show the shape of them.

23. Sketch two different forms of stuffing box, one for high pressure steam, the other for low pressure. Describe how you would pack them and what with.

24. Referring to the drawing example (Figs. 424 to 426).—(*a*) What are the various portions of the bearing made of? Why are the steps in four parts, how are they adjusted when worn, and why is the upper part of the bearing made of cast iron?

25. What conditions make it desirable to use mortise wheels? Give sketches illustrating their construction and proportions. What materials may be used for the cogs?

26. Answer one of the following, but not both :—
(a) Sketch two different forms of stuffing box, one for high pressure steam, the other for low pressure. Describe how you would pack them and what with.
(b) How would you pack the ram of an hydraulic press, and the piston of a double-acting hydraulic cylinder?

27. Sketch a lubricator suitable for (a) the slide bars of an engine, (b) a loose pulley, (c) a line shaft bearing, and describe carefully the action of each.

28. Sketch and describe a form of connecting rod end, either (a) for horizontal engine, (b) locomotive engine, (c) marine engine. How would you adjust the steps when worn so as not to alter the length of the rod?

29. Referring to the drawing examples (Figs. 728 to 735).—Why are the brackets cast separately from the bed plate, how are the brackets lubricated, and what is the object of the groove and oil holes in the brackets? Of what material should each part be made?

30. Sketch two teeth (about 2″ pitch) suitable for a cast iron spur wheel, machine cut. Figure them proportionally, write on names of each part and the curves used.

31. Sketch proportionally (about half size) a single riveted butt joint with double cover plates for ½″ plates. Show plan (about 2 pitches) and sectional elevation, figure it proportionally, and say of what materials each part should be made.

32. Sketch (about full size) a ⅞″ bolt, 3″ long, with a circular head, a bevelled washer and an hexagonal nut. Of what may it be made, and how many vee threads per inch should it have? Make an enlarged view showing proportions and shape of thread, and say how the bolt may be prevented from turning when the nut is put on. Figure the bolt, nut, and washer proportionally.

33. Sketch and describe an adjustable spanner, or a ratchet brace. How would each be used?

34. Make a sketch of a valve suitable for admitting the explosive mixture to the cylinder of a petrol engine.

35. Sketch two teeth (about 2″ pitch) suitable for a cast iron spur wheel, machine cut. Figure them proportionally, write on names of each part and the curves used. How do you measure the diameter at pitch of a toothed wheel?

36. Sketch a form of stuffing box, either for high or low pressure. Describe how you would pack it.

37. Sketch and describe a form of coupling suitable for the main shafting of an engineer's workshop. Why do you recommend the particular one you show?

38. Sketch (about full size) a ¾″ stud, fitted with a capstan-nut and split pin. You may make it any length.

39. Make a hand sketch of a simple form of knuckle joint, and state what part of a steam engine is fitted with this joint.

40. Show by a hand sketch how leather is used to make fluid-tight a spindle for a hydraulic valve.

41. Make a hand sketch of a loose collar, and be careful to show the shape of the end of the set-screw. How should this end be treated to make it effective and durable?

42. Sketch and dimension a cog, suitable for a mortise wheel, the pitch of whose teeth is 3″. Of what wood should such cogs be made, and why?

43. Referring to the drawing example (Fig. 738), of what materials should the piston, gudgeon pin, and set-screws be made? Would you harden any portion of them? If so, how and why? How would you make the piston more pressure tight in the cylinder?

44. Sketch the rim portion of a C. I. spur-wheel showing the teeth about 2″ pitch, give the proportions, names of parts and curves used (a) for ordinary cast gear, (b) for machine cut gear.

45. Sketch and describe either (a) a stuffing box, suitable for a steam pressure of about 50 lbs. per sq. inch, describe how you would pack it and what with; or (b) the packing you would use for an hydraulic ram.

46. How would you joint two cast iron pipes: (a) by a socket joint, or (b) by a flanged joint for a pressure of about 30 lbs. per square inch? Sketch a section of either, describe it and the method of jointing, and materials used.

47. Sketch and describe a connecting rod end suitable for either (a) a stationary engine, (b) a locomotive, (c) a marine engine, or (d) a motor-car engine. How would you adjust it when worn so as not to alter the length of the rod?

48. Show by sketches the sectional shading or lining used to indicate the following materials :—Cast iron, wrought iron, steel, and brass.

49. Show any form of centrifugal lubricator (suitable for an overhung crank pin) that you are acquainted with.

50. Show by sketches three different ways of keying wheels to a shaft, and explain under what conditions each would be used in practice.

51. A single riveted lap joint has ½″ plates, ⅞″ rivets, both steel, f_s and f_t (the ultimate strength in shear and tension) being 23 and 28 tons per sq. inch respectively. Find the most efficient pitch, also the efficiency of the joint.

52. Make neat sketches of the following :—a tee-head bolt, an eye bolt, and a capstan nut.

STAGE 2 (or Second Year's work).

53. Show by sketches and descriptions how you would lubricate any *three* of the following :—(a) slide valve and steam cylinder, (b) overhung crank pin of horizontal engine, (c) crank pin of high speed engine, (d) a loose pulley, (e) an ordinary pedestal, (f) footstep for vertical shaft, (g) an eccentric, (h) the stern thrust bearing of a propeller shaft. State the kind of lubricant you would employ in each case, and how you would ensure its distribution. Name *four* conditions with which a good lubricant should comply.

54. Make a sketch of a vertical section of the half of a mortise mitre wheel, $1\frac{1}{2}''$ pitch. Sketch profile of large end of teeth, showing how they are developed, and give proportions of teeth, method of fixing, and materials employed.

55. Sketch either a double bar guide, or a slipper guide for an engine, and say how you would readjust it when worn. What is the usual pressure allowed per sq. inch of surface on the guide? When is it necessary to use a guide with two surfaces?

56. Give sketches and description of any form of friction gear with which you are acquainted. Show a means of putting it in and out of gear. What are the advantages and disadvantages of this form of gear?

57. Describe briefly the constituent parts and characteristics of cast iron, wrought iron, steel, and give an example of the use of each.

58. How would you join two 4" cast iron pipes (*a*) for a low pressure cold water main, (*b*) for a steam pressure of 60 lbs. per sq. inch, (*c*) for an hydraulic main, 700 lbs. per. sq. inch?

59. Two wrought iron rods, one inch diameter, are to be connected by a cottered joint suitable for alternate compression and tension. Sketch the joint and give the formulæ you would use for calculating the various parts. Where would you prefer it to fail first, and why?

60. In how many ways can a riveted joint fail? Calculate the strength of a double riveted lap joint of $\frac{5}{8}''$ wrought iron plates, $3\frac{3}{4}''$ lap, $\frac{3}{4}''$ rivets pitched diagonally, $1\frac{3}{4}''$ centres. Ultimate strength of plates in tension and rivets in shear 45,000 and 40,000 lbs. per sq. inch respectively. Bearing stress, 90,000 lbs. per sq. inch. Where would the joint fail first?

61. A steam cylinder of cast iron, 10" bore, pressure 70 lbs. per sq. inch, has wrought iron fastenings to the covers, $\frac{3}{4}''$ diameter. How many of them are required (taking $f_t = 3000$ lbs.)? How would you arrange them (*a*) where a port occurred, (*b*) upon the side opposite to a port?

62. Explain how the valve shown in the drawing example (Figs. 354 and 355) acts, and how the various parts of it are made water-tight. How is it connected to the pipe main?

63. Sketch and describe the forms of cranked shafts most suitable for—

(*a*) A horizontal stationary engine.
(*b*) A locomotive engine.
(*c*) A marine engine.

Compare their respective good points. Of what metal should each be made?

64. Sketch and describe a metallic packed stuffing box suitable for a piston rod 4" diameter. How are the whole of the gland studs of a large marine engine tightened by turning one nut?

65. How would you line a large pair of bearing steps with anti-friction metal? What metal would you use for the purpose, and how would you put it in?

66. Sketch and describe joints suitable for a steam pipe, 6" diameter, 80 lbs. pressure; a horizontal pipe for cold water supply, 4" diameter, 30 lbs. pressure; a pipe for cold water, 5" diameter, 700 lbs. pressure. How would you joint each length?

67. Calculate the strength of a mild steel double riveted butt joint with double butt covers. Plates $\frac{5}{8}''$ thick; pitch straight $2\frac{1}{2}''$, diagonally $1\frac{3}{4}''$; lap $3\frac{3}{4}''$; rivets $\frac{3}{4}''$ diameter, pitched diagonally. Where would the joint probably fail, and what would be the percentage of strength compared with solid plate?

68. Sketch and describe another method of fitting the steps in the drawing example (Fig. 424). Of what materials would they be composed?

69. Sketch and describe a removable coupling suitable for the shafting of an engineer's workshop. Of what metal would you make it?

70. Sketch and describe two methods of connecting lengths of cast iron piping for water supply together. What jointing materials would you use? Does the position of the pipe in any way influence your selection of a suitable joint?

71. In how many ways can a riveted joint fail? Write down the formulæ you would use for calculating the tearing, crushing, and shearing of a double riveted lap joint.

72. Sketch and describe (*a*) a gusset stay, (*b*) a tie rod stay, (*c*) a screwed stay suitable for a boiler. In what positions would each be fixed, and what would they be made of?

73. Sketch and describe an end for the connecting rod, suitable for attaching to the cross head shown in the drawing example (Fig. 682).

74. Referring to the drawing example (Fig. 682).—(*a*) What is the object of making the slipper loose, as shown? (*b*) Of what materials should the various parts of the cross head be made? (*c*) What composition would you use for the metal strips, and why are they used? (*d*) The hole in the brasses has worn horizontally $\frac{1}{8}''$ larger on the right and left of the pin: how would you adjust these? (*e*) What is the object of enlarging the bore of the brasses where they come together?

75. A steam cylinder of cast iron, 10 inches bore, pressure 100 lbs. per sq. inch, has wrought iron fastenings, 1" diameter to the covers. How many of them are required (taking $f_t = 3000$ lbs.)? How would you arrange them (*a*) where a port occurred, and (*b*) upon the side opposite to a port?

76. Referring to the drawing example (Figs. 689 and 690).—How would you adjust the brasses when worn $\frac{1}{8}''$ out of truth on the side farthest from the cylinder, and also when worn $\frac{3}{8}''$ in a downward direction? Of what is each portion of the bearing made, and how is it lubricated? What is the object of lipping the cap?

77. Referring to the drawing example (Figs. 728 to 735).—Why are the brackets cast separately from the bed plate? Describe how you would line off and machine the commutator bracket.

78. Sketch and describe a connecting rod little end, *or* a cross head suitable for one of the following:—a horizontal engine, a marine engine, a locomotive engine, a motor-car engine. Describe how you would adjust it when worn, so as not to shorten the rod.

79. Sketch (about half size) and figure proportionally a double riveted (diagonal) butt joint with double cover straps.

80. In how many ways can a riveted joint fail? Write down the formulæ you would use (*a*) in calculating the tearing, crushing, and shearing of a double riveted lap joint, (*b*) in determining the efficiency of the joint.

81. Sketch two teeth (about 2½" pitch) of a spur mortise wheel, give proportions, name of each part, and the curves used. Show how the fixing is accomplished (*a*) into the ordinary rim, (*b*) when occurring over an arm. What material would you recommend for the teeth? Why are mortise wheels now practically obsolete?

82. Sketch and describe a connecting rod big end, suitable for either (*a*) a horizontal engine, (*b*) a marine engine, (*c*) a locomotive, or (*d*) a motor or gas engine. How would you prevent the rod from shortening when adjusting for wear?

83. In how many ways can a riveted joint fail? Write down the formulæ you would use in calculating the tearing, crushing, and shearing of a single riveted butt joint.

84. What limits the size of the rivet for a given thickness of plate?

85. Sketch (about half size) and figure proportionally a double riveted (zig-zag) butt joint with double cover plates. Where would such a joint be used in a cylindrical boiler?

86 A steam cylinder of cast iron, 10" bore, pressure 100 lbs. per sq. inch, has wrought iron studs, ¾" diameter to the covers. How many of them are required (taking f^t = 3000 lbs.)? How would you arrange them (*a*) where a port occurred, and (*b*) upon the side opposite to a port?

87. Sketch and describe a metallic packed stuffing box suitable for a large piston rod. How are the whole of the gland studs tightened by turning one nut?

88. Make a sketch of a valve suitable for admitting the explosive mixture to the cylinder of a petrol engine. What is an automatic inlet valve?

89. Sketch any engine or machine detail in which the adjustment is made by using a gib and cotter. What is the use of the former? When is it necessary to use a set-screw in connection with the cotter?

90. Make a hand sketch of a double riveted zig-zag lap joint for ¾" mild sheet plates. Taking f_t = 28 and f_s = 23 tons per sq. inch, what should the pitch of the rivets be? What efficiency would this joint have? If you have not time to calculate the latter, write down an expression which will represent it.

91. Make a hand sketch of any roller bearing you may be acquainted with.

92. Make a hand sketch of a piston suitable for the cylinder shown on Fig. 628, and give the principal dimensions.

93. If you assume that ¼" has been added to the thickness of the cylinder (Fig. 628) to allow for inequalities of thickness and reboring, what would you estimate the bursting pressure of the cylinder to be? To what pressure would you say it would be safe to work it under steam?

94, A boiler has ⅝" plates which meet, forming a three-plate junction; the joint is made by double butt straps, the circumferential seams being single, and the longitudinal seams double riveted. Make sketches in sectional elevation and plan of joint. Give reasons for your arrangement.

95. How are the tubes put into the tube plates (*a*) of the combustion chamber of a marine boiler, or (*b*) the firebox of a locomotive boiler? Sketch and describe either, giving average sizes and materials used. What is a "Serve" tube and its object?

96. Which do you consider the best method of fixing condenser tubes in place? Sketch it and give usual sizes, and describe its merits and the materials used. Would you put steam or condensing water through the tubes? Give reasons for your preference.

97 In a certain hydraulic press the whole load of 100 tons is taken on two steel bolts, and the working stress at the root section of the threads has been fixed at 6000 lbs. per sq. inch. What size should the bolts be? And what pitch of threads would you recommend? Bearing in mind that the material is steel, would you elect to use plus threads, if so, why?

98. A single riveted lap joint, ½" plates, ¾" rivets, both steel, f_s and f_t the ultimate strength in shear and tension, being 23 and 28 tons per sq. inch respectively. Find the most efficient pitch, also find the efficiency of the joint.

99. The trunnions (or axle ends) of a mixing machine have an effective length of 10", and the weight which comes on each one is 1½ tons. What should their diameter be if the skin stress is not to exceed 5500 lbs. per sq. inch? Note.—In this arrangement you are to assume that the trunnions are only subjected to bending.

100. What are the conditions which allow a wheel to be fixed to a shaft by cone keys? Make a sketch of the arrangement.

101. Make a sketch of a quadruple riveted butt joint, the straps to have scalloped edges. What is the object of giving the edges this form?

102. What advantages have led engineers to use a finer pitch than the standard Whitworth ones for the bolts of cross heads and of connecting rod ends?

103. Calculate the pitch of the studs of a steam cylinder, the diameter of the stud circle being 30", and the number of studs 35. If the diameter of the cylinder at the cover be 25·5", and the steam pressure 80 lbs. per sq. inch, what is the amount of tensional load upon each stud due to steam pressure alone?

Permission to publish the following papers in Machine Construction and Drawing has been kindly given by the Controller of H.M. Stationery Office.

BOARD OF EDUCATION EXAMINATION

SUBJECT II.—MACHINE CONSTRUCTION AND DRAWING (1906)

STAGE 1

Before commencing your work, you must carefully read the following instructions:—

Put the number of the question before your answer.

You are expected to prove your knowledge of machinery, as well as your capability of drawing neatly to scale. You are therefore to supply details omitted in the sketches, to fill in parts left incomplete, and to indicate, by diagonal lines, parts cut by planes of section.

No credit will be given if the candidate shows that he is ignorant of projection. The centre lines should be clearly drawn.

Your answers should be clearly and cleanly drawn in pencil, except the portion specified to be done in ink.

The answers to the questions as well as the drawings should be made on the numbered paper supplied, comprising one sheet of drawing paper with tracing paper and squared foolscap attached. The tracing paper may be detached for the purpose of making the tracing, and must be carefully re-attached.

The value attached to each question is shown in brackets after the question.

You are to confine your answers *strictly* to the questions asked.

The examination in this subject lasts for four hours.

Trace the eye bolt shown on the accompanying Diagram X.

Draw either Example 1 *or* Example 2, Diagram X, but not both. The example should be drawn on the side of the paper on which the candidate's number is printed.

Also answer any two, but not more than two, of the questions numbered 11 to 15.

Tracing, *Diagram X*

Trace in ink on the tracing paper supplied the eye-bolt shown on Diagram X. Insert the dimensions and print the title as shown.
The lines should be very black, of uniform and moderate width, and as continuous as possible. (14)

Example 1, *Diagram X*

$1\frac{1}{4}$-inch Bearing

The diagram gives dimensioned hand sketches of details of a simple bearing. Draw full size, inserting dimensions:—

(*a*) An elevation corresponding with A, but in section, adding the cap and one of the $\frac{5}{8}''$ studs.

(*b*) An elevation, projected from (*a*), looking on the face indicated by the arrow. In this view the cap, cap screws, and the $\frac{5}{8}''$ studs should be shown.

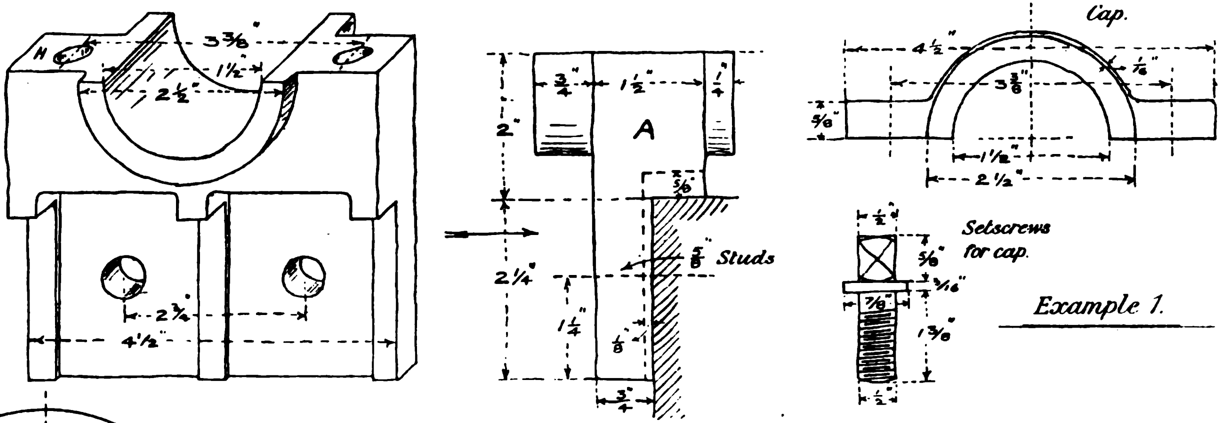
(*c*) A plan.

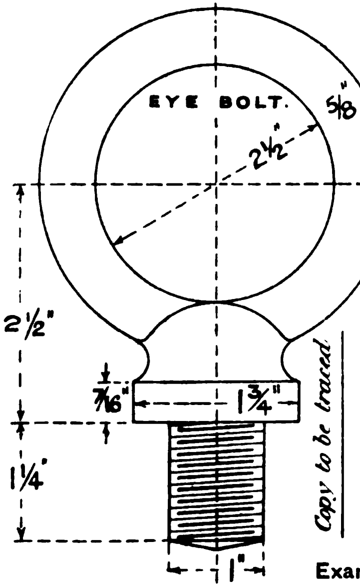
N.B.—Do not draw the pictorial view, nor the parts separated as in the diagram. Dotted lines, representing hidden parts, are not required. (65)

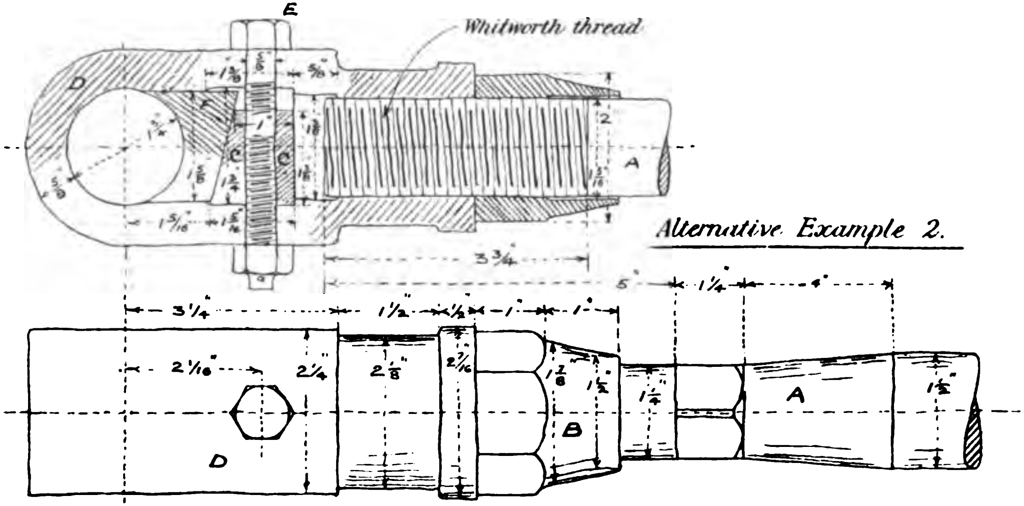
X

SUBJECT II. *Stage 1. 1906.*

NOTE. *Do not draw the views as shown but follow the instructions on the accompanying Examination paper*







Example 1.—BRACKET BEARING. Example 2.—CONNECTING LINK.

Alternative Example 2, Diagram X

End of a Connecting Link of an Air Compressor

Make full size separate scale drawings of details, with dimensions, as follows :—
(*a*) A longitudinal and an end view of the rod end A. The screw thread may be drawn in the manner shown.
(*b*) Three views of the nut B.
(*c*) Three views of the wedge C.
(*d*) Three views of the head D.
N.B.—No credit will be given for drawing the parts assembled, as in the diagram. Dotted lines, representing hidden parts, are not required. (70)

Questions, only two to be answered.

The sketches in answer to these questions should be drawn freehand on the squared foolscap paper the lines on which may be taken as ¼-inch apart.

11. Sketch full size, inserting dimensions, two views of a wheel boss, fixed to a shaft by means of a sunk gib key, as follows :—

Diameter of shaft	2″
Diameter of boss	4″
Length of boss	3″
Width of key	⅝″
Depth of key	⅜″
Taper of key, ⅛″ per foot.	

(8)

12. Name the materials of which the parts of Example 1, and the several parts A, B, C, D, E, and F of Example 2 would be constructed. (8)
13. Sketch full size, inserting dimensions, a 1″ rag bolt *or* Lewis bolt, suitable for securing the frame of a machine to a stone foundation. Explain how the bolt is fixed in the stone. (8)
14. Explain briefly, with sketches, how you would set out, drill, and tap the hole marked H, in Example 1, on the diagram. (8)
15. Sketch in section the armature of a small drum wound motor, showing clearly how the stampings are secured. (8)

Stage 2

Before commencing your work, you must carefully read the following instructions :—

A table of logarithms and functions of angles and useful constants is supplied for each candidate on whose behalf application has been made for a paper in Stage 3 or in Honours.

Put the number of the question before your answer.

You are expected to prove your knowledge of machinery, as well as your capability of drawing neatly to scale. You are therefore to supply details omitted in the sketches, to fill in parts left incomplete, and to indicate, by diagonal lines, parts cut by planes of section.

No credit will be given if the candidate shows that he is ignorant of projection. The centre lines should be clearly drawn.

Your answers should be clearly and cleanly drawn in pencil, except the portion specified to be done in ink.

In Stage 2 the answers to the questions, as well as the drawings, should be made on the numbered paper supplied, comprising one sheet of drawing paper with tracing paper and squared foolscap attached. The tracing paper may be detached for the purpose of making the tracing, and must be carefully re-attached.

The value attached to each question is shown in brackets after the question.

You are to confine your answers *strictly* to the questions asked.

The examination in this subject lasts for four hours.

Trace the **Crank** shown on the accompanying Diagram Z.

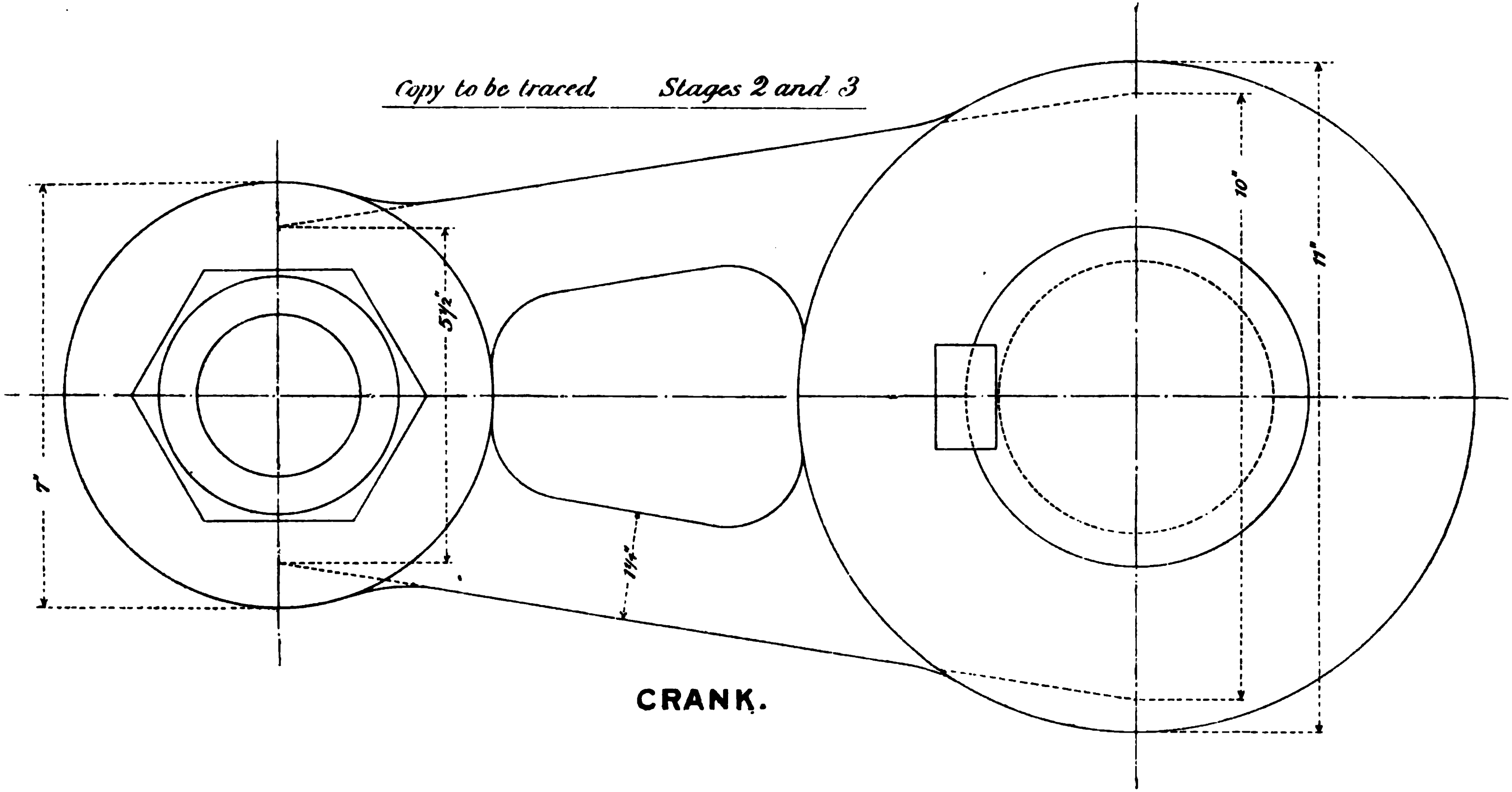
Draw either Example 3 or Example 4, Diagram Y, but not both. The example should be drawn on the side of the paper on which the candidate's number is printed.

Also answer any two, but not more than two, of the questions numbered 21 to 25.

Tracing, *Diagram Z*

Trace in ink, on the tracing paper supplied, the drawing of a crank shown on diagram Z.
The lines should be very black, of uniform and moderate width, and as continuous as possible. (28)

DIAGRAM Z.



CRANK.

Example 3, *Diagram Y*

Adjustable Footstep Bearing

Draw to scale and complete the two projections partly shown in Diagram Y, Example 3, and also draw a vertical section through line *EF*, correctly projected, looking in the direction of the arrow marked *G*.

Scale half size.

No dotted lines need be shown and figured dimensions need not be inserted. (10)

Alternative Example 4, *Diagram Y*

Lift Valve

Draw, full size, an outside view corresponding to the vertical section shown in Example 4, Diagram Y. Draw also a sectional plan through line *CD*, with the spindle *K* removed. Finally draw a sectional elevation taken through the centre line, looking in the direction of the arrow marked *H*.

Also make a plan and the two end elevations of the spindle marked *K*, all the necessary dimensions being shown on it.

Scale full size.

No dotted lines need be shown and figured dimensions need not be inserted in the three first views. (140)

Questions, only two to be answered

The sketches in answer to these questions should be drawn freehand on the squared foolscap paper

21. State of what material you would make the parts marked *M*, *N*, *O*, *P*, in the footstep drawing, Diagram Y, Example 3. Also sketch an arrangement to prevent the rotation of the footstep bearing (*N*) in the casting *O*. (16)

22. A wrought iron crank shaft is formed by bending a 2″ round bar. How would you proceed to turn the crank pin? (16)

23. A plate girder is made up of a vertical web 2′ deep connected to top and bottom flanges, 1′ wide, by two angle irons 3″ × 3″ × ½″. The web and flange plates are ½″ thick. Sketch a section of the above girder, putting in the necessary dimensions. Show also a suitable stiffener. (16)

24. Sketch to scale, half size, inserting dimensions, a double riveted butt joint with two straps, as used in the longitudinal joint of a boiler. Rivets ⅞″ diameter, pitch 3″, and thickness of plate ¾″. What would be the efficiency of this joint? (16)

25. Show, by sketches, the method of holding and insulating the bars of a commutator of a continuous current dynamo. The shaft is 3″ diameter, and the outside of the bars 8″ diameter. (16)

Stage 1.—(1908)

Trace the copy shown on the accompanying Diagram X.

Draw either Example 1 *or* Alternative Example 2, Diagram X, but not both. The example should be drawn on the side of the paper on which the candidate's number is printed.

Also answer any two, but not more than two, of the questions numbered 11 to 15.

Tracing, *Diagram X*

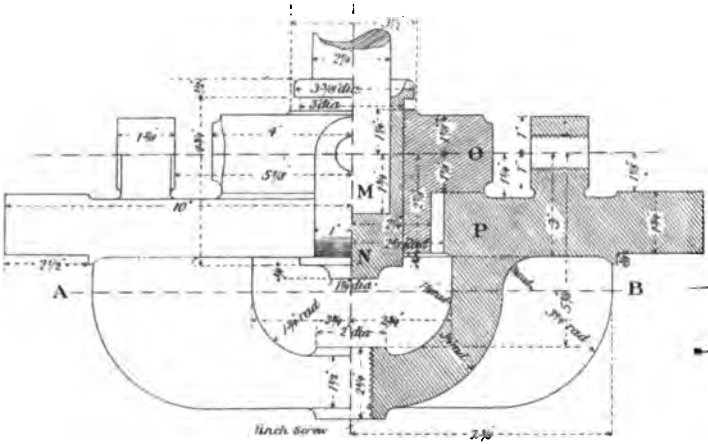
Trace in ink on the tracing paper supplied the two views of the eye bar shown on Diagram X. Insert the dimensions and print the title as shown.

The lines should be very black, of uniform and moderate width and as continuous as possible. (14)

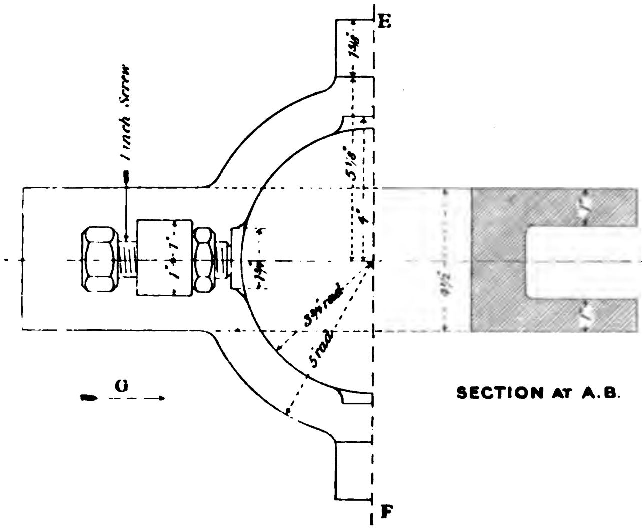
DIAGRAM Y. SUBJECT III, STAGE 2, 1906.

NOTE.—Do not draw the views as shown, but follow the instructions on the accompanying Examination Paper.

ADJUSTABLE FOOTSTEP BEARING.

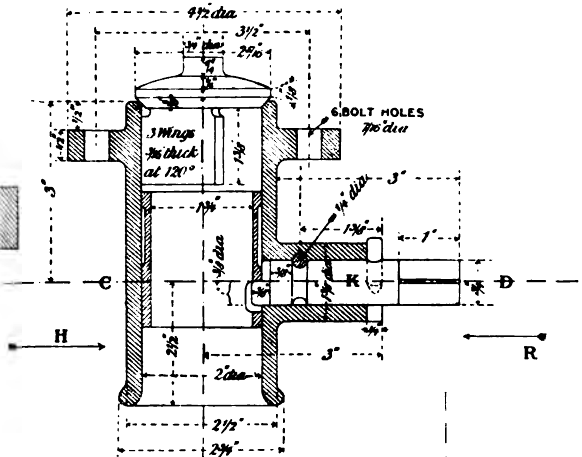


Example 3. Draw ½ size.



SECTION AT A.B.

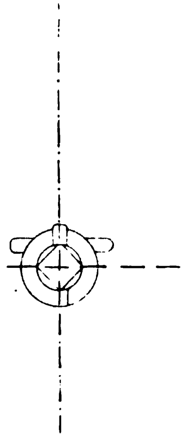
LIFT VALVE.



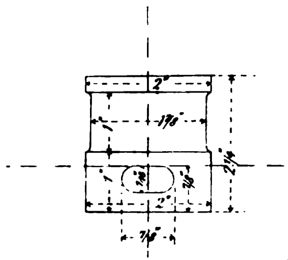
VERTICAL SECTION

Alternative Example 4.

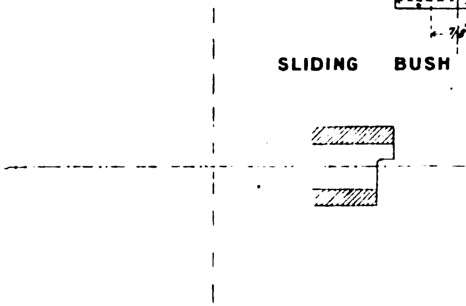
Draw full size.



PART OF OUTSIDE ELEVATION LOOKING IN THE DIRECTION OF THE ARROW R



SLIDING BUSH FOR LIFTING VALVE



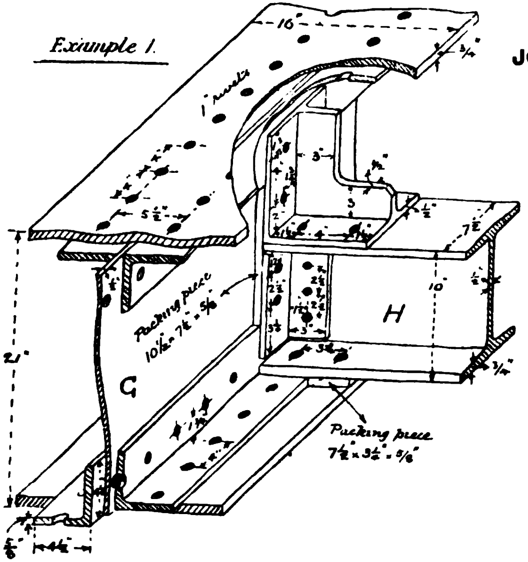
PART OF SECTIONAL PLAN THROUGH C.D.

DIAGRAM X. SUBJECT II, STAGE I. 1908.

NOTE.—Do not draw Example 1 or 2 as shown, but follow the instructions on the accompanying Examination Paper.

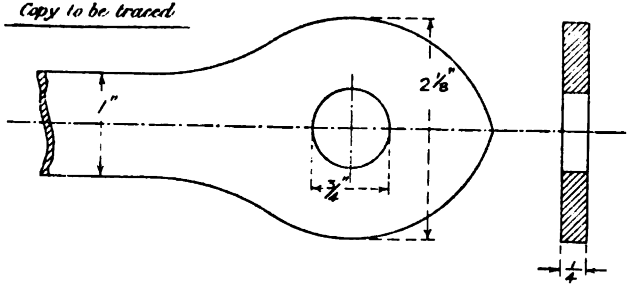
Example 1.

JOINT IN GIRDER WORK.



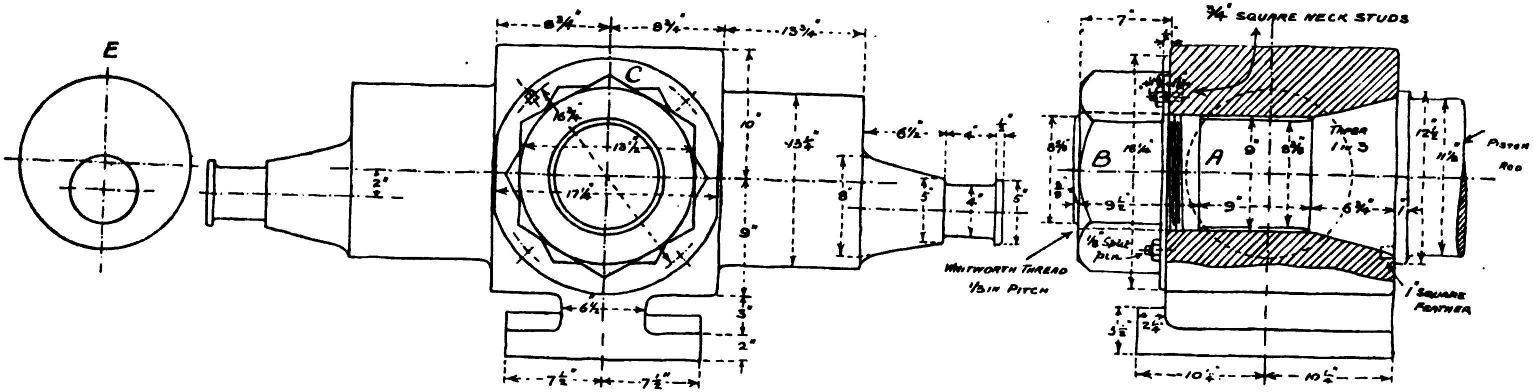
EYE BAR

Copy to be traced



Alternative

Example 2.



PISTON-ROD END AND CROSS HEAD.

Example 1, Diagram X

Joint in Girder Work

A girder *G*, 21" deep, is built up of 16" × ¾" flange plates, ½" web, 4½" × 4½" × ⅝" angles, and 1" rivets of 4" pitch. A cross girder *H*, 10" deep, of rolled joist section, with 7½" × ¾" flanges and ½" web, rests on the lower flange of *G*, and is riveted to *G* by means of two 3½" × 3½" × ½" angle pieces, each 7½" long, and a bent 7½" × 3½" × ½" tee piece, as shown in the dimensioned pictorial sketch.

Draw, to a scale of ¼, inserting dimensions, two elevations and a sectional plan of the joint, the horizontal section plane being taken through the axis of the cross girder *H*.

N.B.—Do not draw a pictorial view after the manner of the diagram. (70)

Alternative Example 2, Diagram X

Piston-Rod End and Cross Head

Make separate scale drawings of details, inserting dimensions, as follows:—

(*a*) Two views of the piston-rod-end *A*. The screw thread may be represented conventionally as in the diagram. Scale ¼.
(*b*) Three views of the nut *B*. Scale ¼.
(*c*) Two views of the locking plate *C*. Scale ¼.
(*d*) Three views of one of the ¾" square-neck studs, with nut and split pin in position. Scale full size.
(*e*) Complete the end elevation *E* of the cross head, with the piston rod, nut, and locking plate removed. Scale ¼.

N.B.—Do not draw the other views of the cross head, nor the parts assembled as in the diagram. Dotted lines, representing hidden parts, are not required. (70)

Questions, only two to be answered

The Sketches in answer to these questions should be drawn freehand on the foolscap paper, which is ruled in ¼-inch and ⅛-inch squares (8)

11. State the use of the locking plate *C* of Example 2, and describe how it is got into position and fixed.

12. Referring to Example 2, what purpose is served by the square neck on the ¾" stud, and by the ⅛" split pin? Sketch the latter. What is the object of using the 1" square feather? (8)

13. Sketch in section, half size, inserting dimensions, a flange joint for a 5" cast-iron steam pipe, secured by six ¾" bolts, to the following dimensions:—

Thickness of metal of pipe	⅝"
Diameter of flange	10"
Thickness of flange	13/16"
Radius of bolt circle	4⅛"

State how the joint is made steam-tight. (8)

14. Sketch in longitudinal section, full size, a simple gland and stuffing box suitable for a 1" piston rod, there being two ⅝" gland studs, 2¾" apart from centre to centre, the diameter of the gland being 1⅝", and the internal depth of the stuffing box 1½". Insert dimensions. Describe how the packing is put in place. (8)

15. Sketch in detail the insulator, and the method of fixing, for carrying an ordinary telegraph wire. (8)

Stage 2, Diagram Y

Read the General Instructions on page 1.

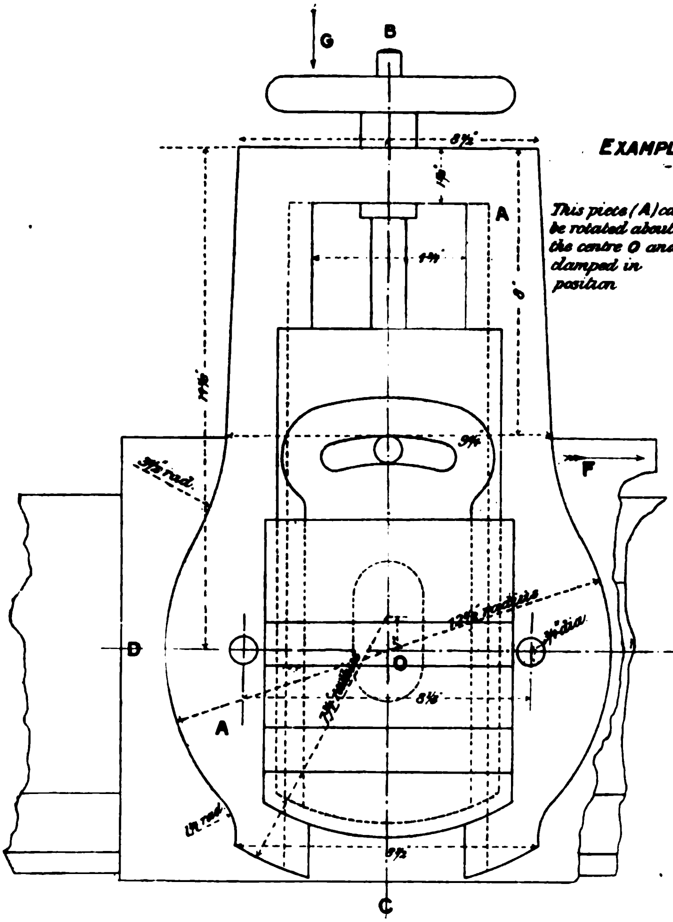
Trace the half fly-wheel pulley shown on the accompanying Diagram Y.

Draw either Example 3 or *alternative* Example 4, Diagram Y, but not both. The example should be drawn on the side of the paper on which the candidate's number is printed.

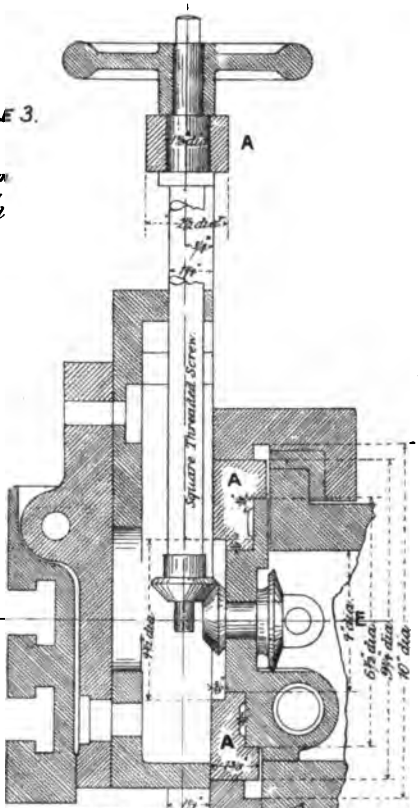
Also answer any two, but not more than two, of the questions numbered 21 to 25.

TOOL HOLDER FOR A PLANING MACHINE.

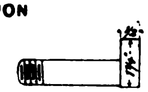
EXAMPLE 3.



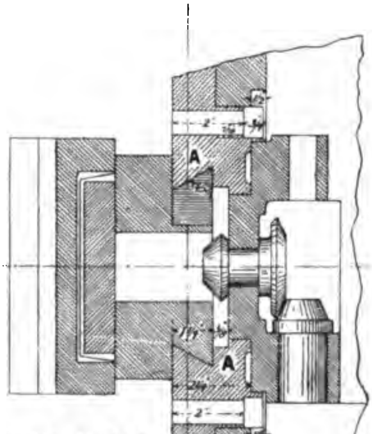
OUTSIDE ELEVATION LOOKING IN THE DIRECTION OF THE ARROW F.



SECTION THROUGH CENTRE LINE B.C.



SQUARE HEADED BOLT FOR CLAMPING A.

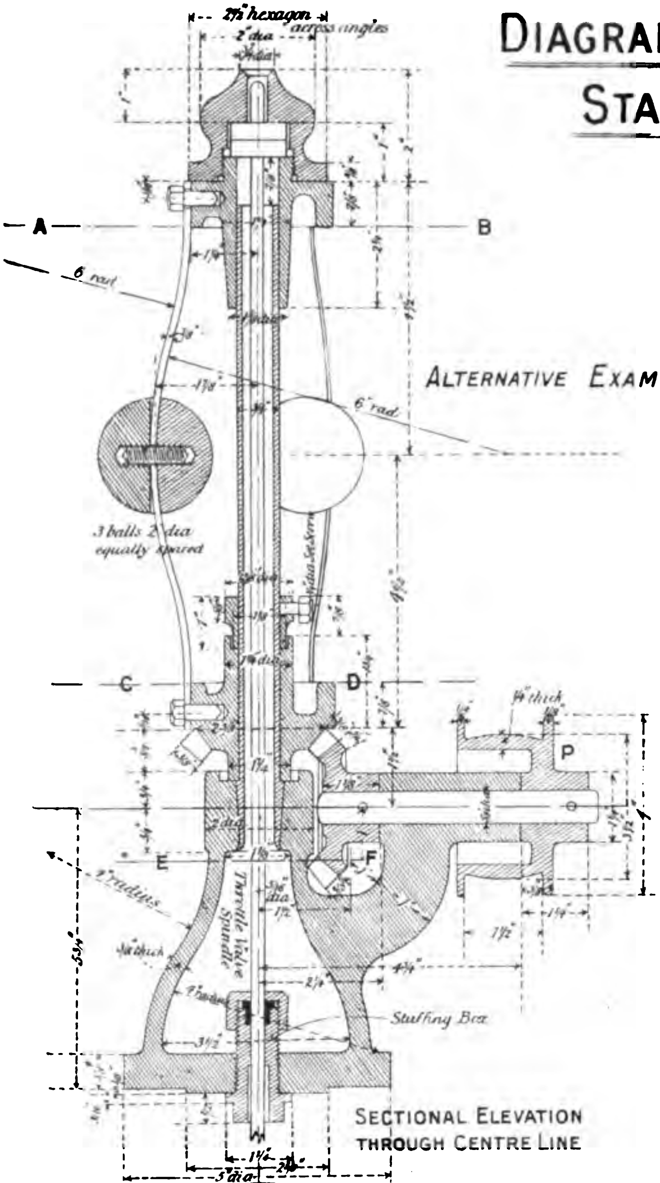


SECTIONAL PLAN THROUGH D.E.

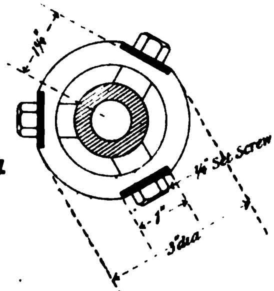
DIAGRAM, Y. SUBJECT II, STAGE 2, AND STAGE 3, 1908.

NOTE – Do not draw the views as shown, but follow the instructions on the accompanying Examination Paper

Diagram, Y. Subject II, Stage 2, 1908.

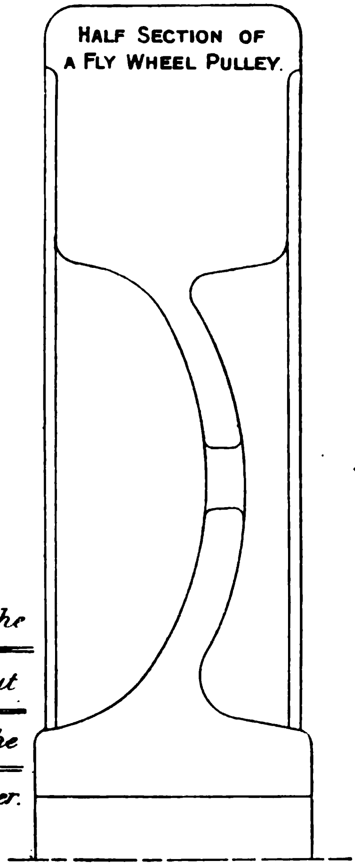


Alternative Example 4



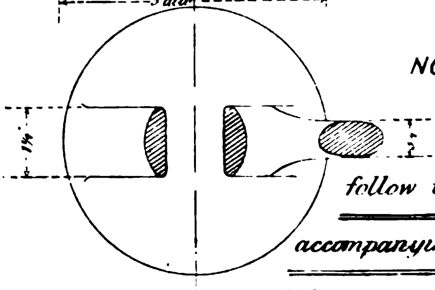
Section through A.B. looking up and through C.D. looking down.

Drawing to be traced Stages 2 & 3.



Sectional Elevation through Centre Line

NOTE ~ Do not draw the views as shown, but follow the instructions on the accompanying Examination Paper.



Section through E.F.

A STEAM ENGINE GOVERNOR.

Tracing

Trace in ink, on the tracing paper supplied, the half fly-wheel pulley shown on Diagram Y. and headed "Drawing to be traced." The lines of the tracing should be black, uniform in width, and of moderate thickness. (28)

Example 3, *Diagram Y*

Tool Holder for a Planing Machine

Draw the following views of the part of the Planing Machine Tool Holder marked *A* to a scale one-half full size :—

(*a*) A view of *A* corresponding to the outside elevation.
(*b*) A half plan and half section of *A* through the line *DE* looking in the direction of the arrow *G*.
(*c*) A section of *A* through the centre line *BC*.

Note.—No other part of the holder is to be drawn but the part *A*.
Neither dotted lines nor dimensions need be shown. (140)

Alternative Example 4, *Diagram Y*

A Steam Engine Governor

Draw an outside elevation corresponding to the sectional elevation shown in the diagram, omitting the stuffing box, and a complete plan.
Scale ¾ full size.
The mitre wheels may be shown as rolling cones.
Neither dotted lines nor dimensions need be shown. (140)

Questions, only two to be answered

The sketches in answer to these questions should be drawn freehand (either in pencil or in ink), by the side of the written answer on the squared foolscap paper. The sketches must not be made on the drawing paper.

21. Sketch a flange joint suitable for connecting two 4″ steam pipes carrying steam at a pressure of 120 lbs. per sq. inch. (16)
22. Make a sketch of a gusset stay suitable for staying the flat end of a Lancashire boiler. (16)
23. Sketch a pattern suitable for the moulding of the pulley *P* for the Governor, Alternative Example 4, Diagram Y, and briefly describe how you intend that the pulley shall be moulded. (16)
24. Make a sketch of an eccentric sheave and show clearly how it is secured to the shaft.

Sketch an end view of the shaft, and on it show the angular advance of the sheave and its radius or eccentricity, the crank being indicated by a dotted line, and the direction of rotation by an arrow. It is assumed that the eccentric is driving an ordinary slide valve. (16)
25. Sketch a section of the armature of a small continuous current motor, showing clearly the construction of the commutator and the armature core. You are not expected to show any of the windings, but only the mechanical details of the type of armature you select for description. (16)

CITY AND GUILDS OF LONDON INSTITUTE

TECHNOLOGICAL EXAMINATIONS

MECHANICAL ENGINEERING (1906)

ORDINARY GRADE.—PART II. (SECOND YEAR'S COURSE)

INSTRUCTIONS

If the Candidate has already passed in this subject, in the first class of the Ordinary Grade, he cannot be re-examined in that grade.
The Candidate must state on the top of his answer paper the Section in which he is examined, and must select his questions from *one Section only*.
Candidates in any one of the Sections B, C, D, or E are required to do the drawing of the **Bearing Bracket**, Figs. 1 and 2.
The maximum number of marks obtainable is affixed to each question.
The number of the question must be placed before the answer in the worked paper.
A sheet of drawing paper is supplied to each Candidate. The Candidate is at liberty to use divided scales, compasses, set-squares, and the slide rule.
Three hours allowed for this paper.

SECTION A.—MACHINE DRAWING

Candidates in this Section only are allowed the use of any one pocket-book or treatise on machine designing; but the title of the pocket-book or treatise used must be stated at the head of the answer paper.

N.B.—To obtain full marks for a drawing, it must be fully dimensioned, and the materials of which the parts are made indicated by sectional shading. Choose example 1 *or* 2. *The figures are given on the plates attached. They are not drawn to scale.*

1. Fig. 3 (Plate A) is the sectional elevation of a pump; Fig. 4 is an incomplete end elevation; Fig. 5 is a sectional plan with the valve removed; and Fig. 6 shows the details of the valves and seatings. In place of Fig. 3 draw an elevation of the pump; in place of Fig. 4 a sectional end elevation on the plane CD; and in place of Fig. 5 a plan of the pump. The views shown are not to be drawn. Scale, ¼ full size. Add any omitted detail. (100 marks)

or

2. Figs. 7 and 8 (Plate B) show two views of an eccentric sheave and strap. Complete the elevation, Fig. 7, adding any omitted detail, and showing the strap connected to the end of the eccentric rod. In place of Fig. 8, draw a section on the plane AB, and add a complete plan. Scale ¼ full size. (100)

DRAWING EXAMINATION FOR CANDIDATES IN SECTIONS B, C, D, AND E

No pocket-books allowed

In Figs. 1 and 2 are shown two views of a bearing bracket. Draw the elevation, Fig. 1, a section on CD, looking in the direction of the arrow, and a sectional plan on the plane AB. Do not draw Fig. 2. Scale full size. (20 marks)

SECTION B.—PATTERN MAKING

Not more than four questions are to be attempted in addition to the drawing of the bearing bracket shown in Figs. 1 *and* 2.
More marks are given for neat sketches than for vague and general descriptions.

1. Show, with the aid of sketches, how you would make patterns and coreboxes for the pump case shown in Figs. 3–6 (Plate A). What kind of timber would you use? (20)
2. A fly-wheel, about 5 ft. diameter, having elliptical arms, a rim of rectangular section, and a circular boss, is to be moulded. Sketch the patterns, striking boards, and coreboxes, you would supply. (20)
3. Show, in detail, how you would make the pattern and core box for the bearing bracket shown in Figs. 1 and 2. (20)

4. Make sketches showing the construction of the patterns for plate moulding any three different articles which are to be produced in large quantities. (20)

5. What is meant by a "Boxed-up Pattern"? Describe the construction of a boxed-up pattern for a machine frame, such as, for instance, the frame of a pattern shop planing machine or band saw, or of any such pattern you have worked on. (20)

6. Give sketches showing examples of—
   (a) Building up a pattern.
   (b) The construction of bends and corners.
   (c) Two forms of halvings.
   (d) A method of fixing a curved flat piece to a straight flat piece. (20)

N.B.—*All Candidates are required to draw the bearing bracket, Figs.* 1 *and* 2, *in the way described.*

BEARING BRACKET.

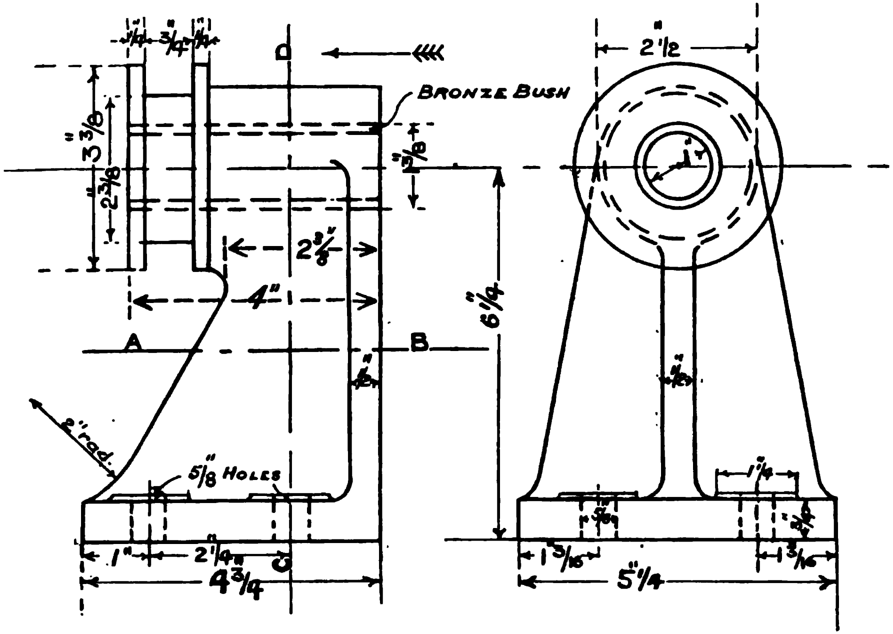


Fig. 1. Fig. 2.

Section C.—Foundry Work

Not more than four questions are to be attempted in addition to the drawing of the bearing bracket shown in Figs. 1 *and* 2.

More marks are given for neat sketches than for vague and general descriptions.

1. Make sketches showing the construction of the mould for the pump casing shown in Figs. 3–5 (Plate A). The gates, and the way in which the cores are supported, are to be clearly indicated. (20)

2. Describe, with the aid of sketches, the method of making a loam mould of either—
   (a) A chain barrel, 10" diameter, 30" long, for a crane, with a helical groove for the chain.
   (b) A large speed cone having four steps varying in diameter from 18" to 3'. (20)

3. Sketch some form of crucible furnace suitable for a brass foundry, and describe the preparation of alloys suitable for—
   (a) The valves of the pump shown in Figs. 3–6.
   (b) The brass bush of the bearing shown in Figs. 1 and 2.
   (c) Small brass cocks. (20)

4. Sketch and describe a cupola suitable for an iron foundry, and describe the charging of the cupola, giving the relative quantities of fuel and iron. The iron produced should be suitable for the pump casing of Question 1. (20)

5. Describe the method of making, and state the materials used in the construction of the following cores:—
   (a) For the pump case shown in Figs. 3–6.
   (b) For the bearing shown in Figs. 1 and 2.
   (c) The cores for the ports of a steam engine cylinder.
   (d) For a pipe or hollow column.
   Particular attention should be paid to the venting of the cores. (20)

6. Give sketches of three articles, and the patterns of them, for which plate moulding would be suitable. Describe the method of making the moulds. (20)

7. Make a sketch of a mould for a tram wheel, the tread of which is to be chilled. Mention any special precautions that have to be taken in making chilled castings. (20)

N.B.—*All Candidates are required to draw the bearing bracket, Figs.* 1 *and* 2, *in the way described.*

PUMP.

FIG 3.

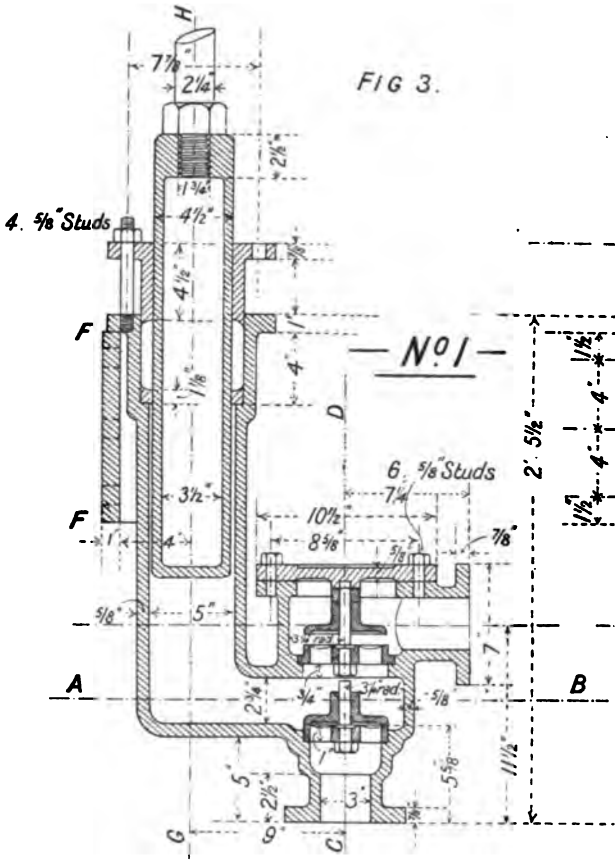


FIG 4.

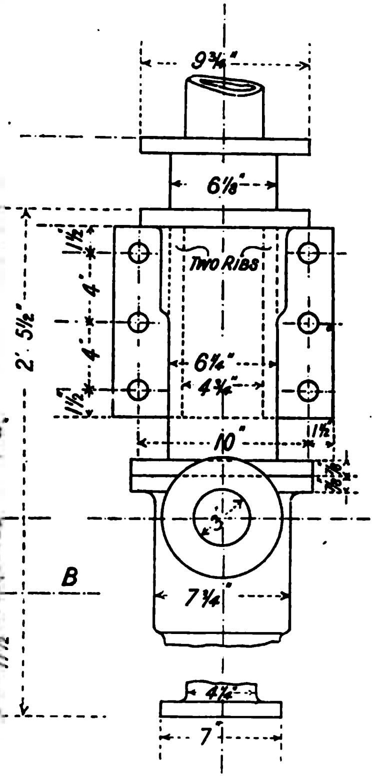
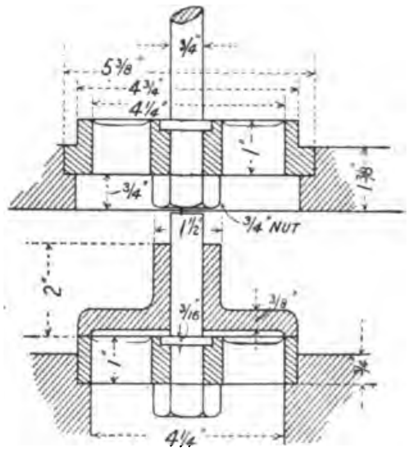
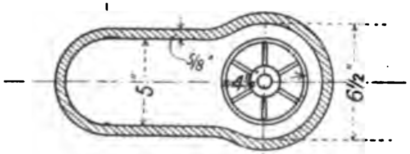


FIG 6.





— SECTION ON A.B. —
FIG 5.

PLATE A.

ECCENTRIC SHEAVE AND STRAP.

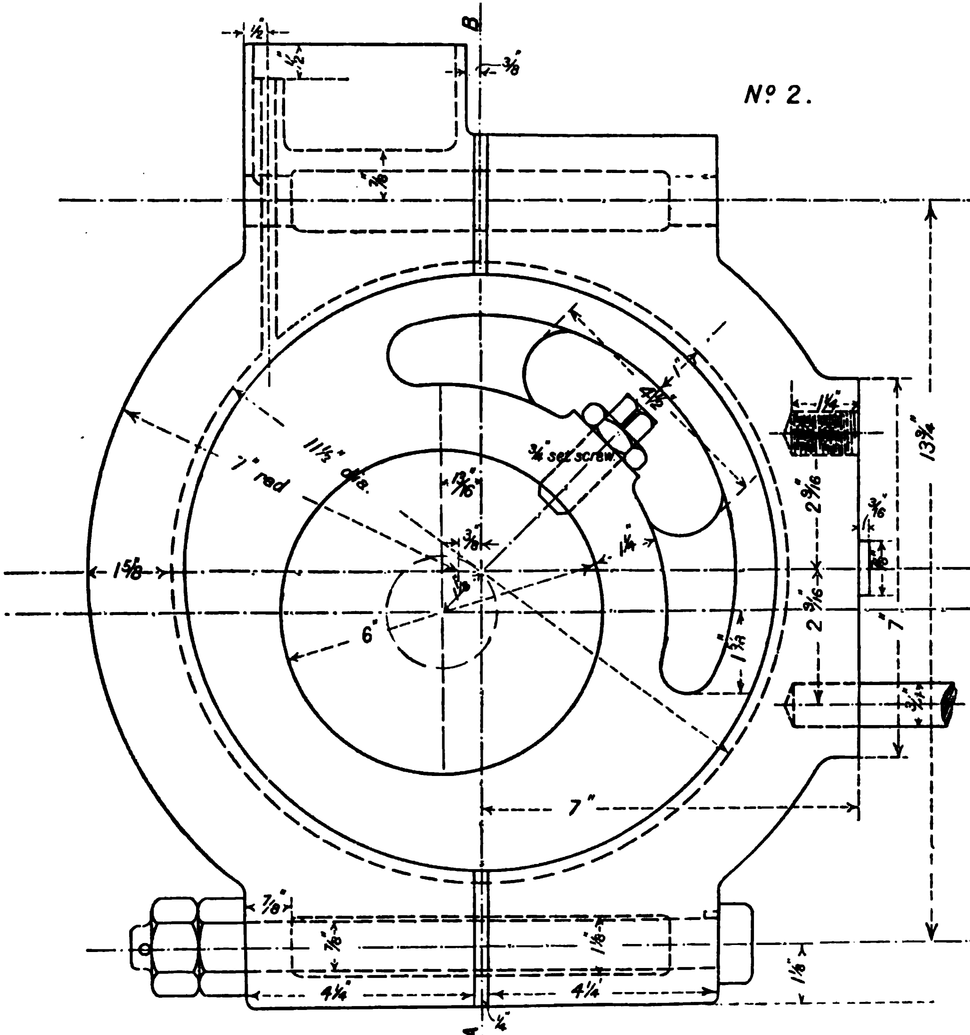


FIG. 7.

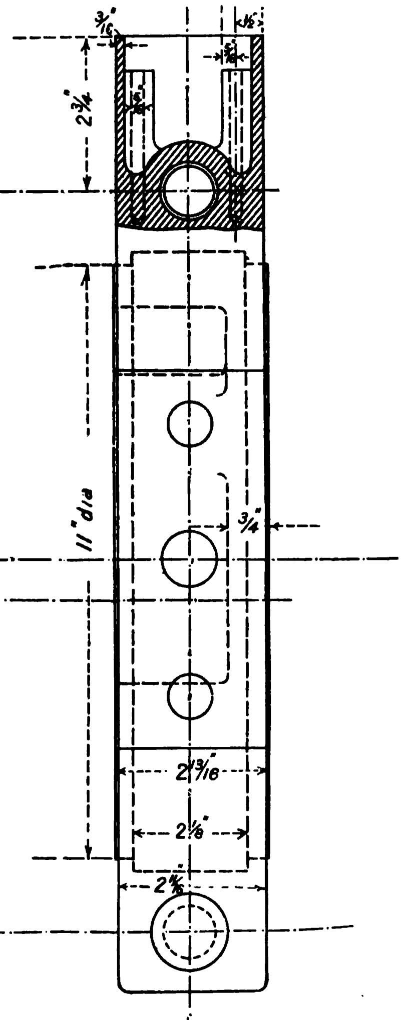


FIG. 8

PLATE B.

Section D.—Fitters' and Turners' Work

Not more than four questions are to be attempted in addition to the drawing of the bearing bracket shown in Figs. 1 *and* 2.
More marks are given for neat sketches than for vague and general descriptions.

1. The casting for the pump shown in Figs. 3-6 (Plate A) has to be bored for the valve seatings and in the gland box, and machined on the face FF. The bores must be parallel to each other, and to the face FF. Describe carefully the treatment of the casting in the fitting shop, showing, by the aid of sketches, how you would machine the face AB, and bore the casting. (20)
2. Suppose the valves of the pump shown in Figs. 3-6 to be leaking, how would you true up the valves and seatings? How would you make the joint of the valve box cover, and what kind of packing would you use in the stuffing box? (20)
3. Describe carefully and show by sketches the processes of machining and turning the eccentric sheave and strap shown in Figs. 7 and 8 (Plate B). (20)
4. Make sketches of limit gauges suitable for gauging to within $\frac{1}{1000}$ in. :—
   (*a*) The diameter of cranks or journals.
   (*b*) The diameter of cylinders, about 8" diameter. (20)
5. The diameters of the steps of the cone pulleys of a drilling machine are 6", 7¼", 9", and 10¼", and the countershaft cone pulley is similar. The countershaft runs at 320 revolutions per minute. Find the maximum and minimum number of revolutions of the cone pulley of the machine. Sketch an arrangement of pulleys on the countershaft so that the motion of the machine may be reversed. For what purpose is it required to reverse a drilling machine? (20)
6. What is the object of chipping strips? Make a sketch of an ordinary pedestal, and describe the fitting of the brasses into the body and cap. Also describe the boring of the brasses and the arrangement of the oil grooves. (20)
7. The pump, Figs. 3-5 (Plate A), the eccentric sheave and strap, Figs. 7 and 8 (Plate B), the forging of the lever, Fig. 9, which is to be finished bright all over, and the bearing, Figs. 1 and 2, are sent to the marking-off table to be prepared for machining. Describe the marking off, and the tools you would use. (20)

N.B.—*All candidates are required to draw the bearing bracket, Figs.* 1 *and* 2, *in the way described.*

LEVER.

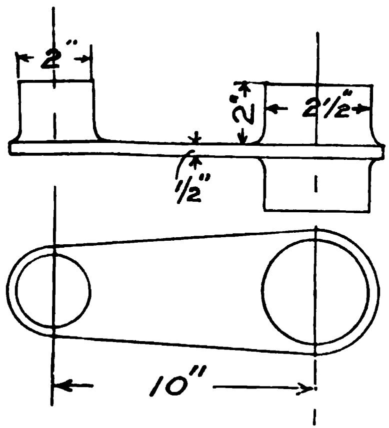


Fig. 9.

Section E.—Smiths' Work

Not more than four questions are to be attempted in addition to the drawing of the bearing bracket shown in Figs. 1 *and* 2.
More marks are given for neat sketches than for vague and general descriptions.

1. Describe the method of forging a steel double-throw locomotive crank shaft with cranks at right angles. (20)
2. What kind of material would you use for a crane chain? Describe the forging of the chain, mention two kinds of defective welds in chains, and state what precautions you would take to avoid them. Explain in detail how the chain should be treated after leaving the anvil, before being used. What is the object of this treatment? (20)
3. Make sketches of a reverberatory regenerating gas furnace, suitable for heating the crank of Question 1. (20)
4. Describe carefully how you would forge the lever shown in Fig. 9 from the solid. (20)
5. Make sketches of two kinds of tongues which are frequently used by a smith, and explain in detail how you would make them. (20)
6. Describe the process of hardening and tempering—
   (*a*) A chisel for cutting mild steel.
   (*b*) A screw tap.
   (*c*) A spiral spring.
   Also explain how you would case-harden—
   (*d*) A crank pin, 6" diameter.
   (*e*) A number of small articles. (20)

N.B.—*All Candidates are required to draw the bearing bracket, Figs.* 1 *and* 2, *in the way described.*

MECHANICAL ENGINEERING (1908)

Ordinary Grade.—Part II. (Second Year's Course)

Section A.—Machine Drawing

Candidates in this Section only are allowed the use of any one pocket-book or treatise on machine designing; but the title of the pocket-book or treatise used must be stated at the head of the answer paper.

N.B.—To obtain full marks for a drawing, it must be fully dimensioned, all views must be correctly projected, and the materials of which the parts are made indicated by sectional shading. Choose example 1 *or* 2. *The figures are given on the plates attached. They are not drawn to scale. The drawings need not be inked in.*

1. Figs. 1 to 4 (Plate A) are incomplete views of a ring bearing. Draw to a scale of $\frac{3}{4}$ full size:—

(1) An elevation, looking in the direction of arrow 1, Fig. 2.

(2) A complete plan under view (1), the upper half showing the cap only removed.

(3) A half section on the plane GH, looking upwards

(4) A half section on the plane KLM.

Do not draw the views as shown on the paper. Add omitted detail. (100 marks)

or

2. Figs. 5 to 10 (Plate B), show incomplete separated details of an adjustable cross head for a horizontal steam engine. Draw to a scale of $\frac{3}{4}$ full size:—

(1) A section on the plane AB of the complete cross head.

(2) An elevation, looking in the direction of the arrow 1, Fig. 7.

(3) A section on the plane CD, looking in the direction of the arrow 2, Fig. 7.

(4) A half plan, and a half section on the plane EF.

The views shown on the plate are not to be drawn. Add omitted detail, especially showing how the gudgeon pin is prevented from rotating. (100)

Drawing Examination for Candidates in Sections B, C, D, and E

No pocket-books allowed

Figs. 5 and 6, Plate B, show details of the slipper of a cross head. Draw to a scale of full size:—

(1) A plan, looking on the top of the slipper. (20)

(2) A plan, looking underneath the slipper. (20)

(3) A section on the plane GH, looking in the direction of arrow 3, Fig. 5. (20)

RING BEARING.

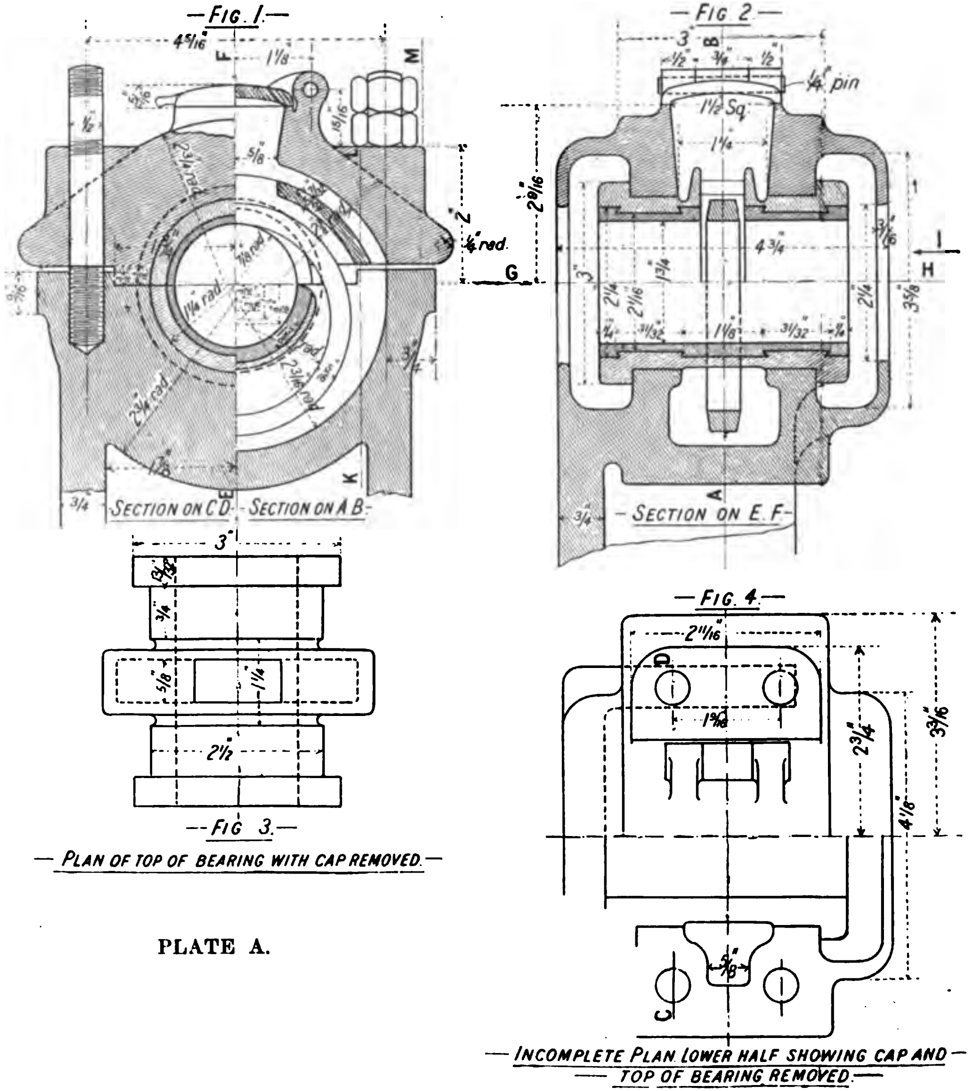


PLATE A.

CROSS HEAD FOR HORIZONTAL STEAM ENGINE.

Details of Slipper

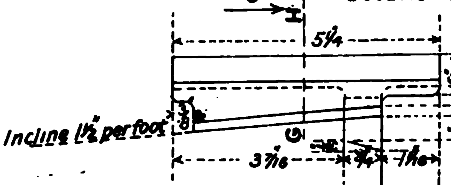


FIG. 5.

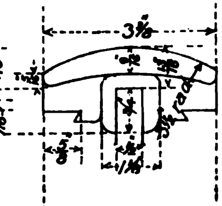


FIG. 6.

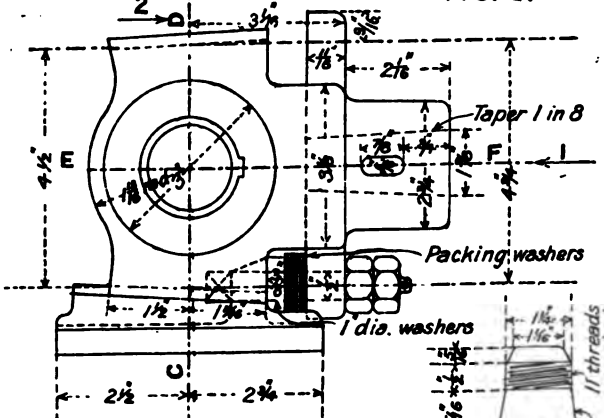
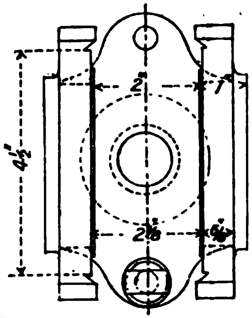
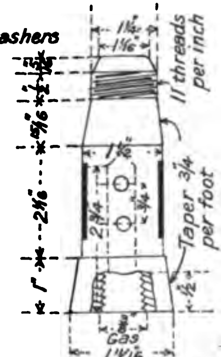


FIG 7.
Elevation with one Slipper removed

FIG 8.
End Elevation with Slippers removed





Detail of gudgeon pin

FIG. 10.

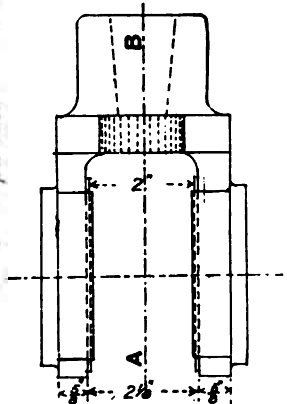


FIG 9.
Plan with Slippers removed.

PLATE B.

SECTION B.—PATTERN MAKING

Not more than four questions are to be attempted in addition to the drawing shown in Figs. 5 and 6, Plate B (see above).

More marks are given for neat sketches than for vague and general descriptions.

To obtain full marks it is necessary that the detailed construction of the patterns should be shown by neat sketches.

1. Describe, with the aid of sketches, how you would make the pattern of a worm, having four complete threads of 1¼" pitch, the outside diameter of which is 6"; the worm to have an axial hole, to be bored 2" diameter. (20)

2. Show, by sketches, the construction of a band saw, and how the band is kept tight and is prevented from twisting. Why are the pulleys of the band saw generally made of light construction? Describe briefly any other machines found in a good pattern shop. (20)

3. What do you mean by a framed pattern? Show in detail how you would make a framed pattern for a large flat plate, and describe either the construction, in detail, of a framed pattern for a bracket such as is used for carrying a shaft pedestal, or else describe the construction of a framed pattern on which you have worked. (20)

4. Show in detail, and describe step by step, how you would make the pattern and core boxes, if any, of the cast-iron cap of the bearing shown in Fig. 1, Plate A. Put six dimensions on the sketches to show the actual dimensions you would make the pattern to allow for contraction. (20)

5. Make detailed sketches showing how you would make the pattern of the part of the cross head shown in Plate B which is cottered to the piston rod. Put on a few over-all dimensions to show the allowance for contraction. (20)

6. Show how you would construct skeleton patterns for a circular domed cover, as shown in Fig. 11; to be cast in green sand. Sketch also the strickle boards that would be required. (20)

N.B.—*All Candidates are required to draw the slipper in the way described.*

SECTION C.—FOUNDRY WORK

Not more than four questions are to be attempted in addition to the drawing shown in Figs. 5 *and* 6, *Plate B.*
More marks are given for neat sketches than for vague and general descriptions.

1. Describe the moulding, and make sketches of the moulds, of the cap of the bearing shown in Plate A. What would be the probable mixture of the iron you would use for such a casting? (20)
2. Show by sketches, and explain briefly, how you would mould either a pipe, 12″ diameter and 10′ long, with ordinary flanges, or (2) a hydraulic crane cylinder suitable for a ram about 9″ diameter, having a stroke of 9′. (20)
3. Make sketches of a furnace suitable for an iron foundry. Describe the charging for a blow suitable for casting steam engine cylinders. (20)
4. A moderate sized fly wheel is to be cast with a split boss. Describe, with sketches, the preparation of the mould and cores. (20)
5. Make sketches of a moulding machine suitable for moulding toothed wheels. (20)
6. Sketch in detail any mould upon which you have recently worked in the foundry. (20)
7. Make sketches of a mechanical tipping ladle suitable for cast iron. Give the composition of the lining, and show carefully how the plug is worked. (20)

N.B.—*All Candidates are required to draw the slipper, Figs.* 5 *and* 6, *Plate B, in the way described.*

CIRCULAR DOMED COVER.

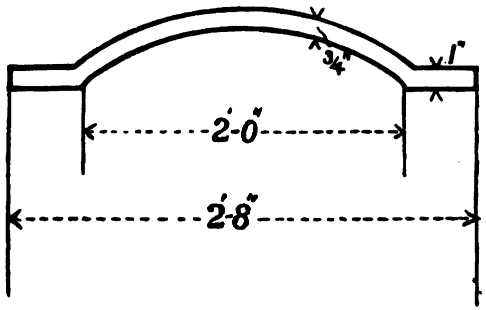


FIG. 11.

SECTION D.—FITTERS' AND TURNERS' WORK

Not more than four questions are to be attempted in addition to the drawing shown in Figs 5 *and* 6, *Plate B.*
More marks are given for neat sketches than for vague and general descriptions.

1. Describe, with sketches, the marking-off, and the machining and turning of the central part of the cross head (not including the gudgeon pin) shown in Figs. 7 to 9, Plate B. (20)
2. Describe carefully, and illustrate with sketches, how you would turn the gudgeon pin shown in Fig. 10, Plate B, and calculate a suitable train for wheels cutting the thread, assuming the leading screw of the lathe has four threads to the inch. (20)
3. The castings for the bearing shown on Plate A are received in the fitting and turning shop. Show, with sketches, and describe briefly how you would treat them to obtain a true gearing. (20)
4. Show how you would set the D slide valve of a horizontal steam engine. (20)
5. A job, 24″ diameter, which can be fixed to a face plate, is to be turned, and the largest lathe in the shop is a 10″ centre lathe. Show, by sketches, how you adapt the lathe so that it can turn the job. Sketch in detail any new parts you would require. (20)
6. A line of shafting, 3″ diameter, is to be carried on bearings on brackets fixed to a wall. Describe how you would set the pedestals on the brackets so that they shall all be truly in line. Sketch one of the brackets and a pedestal. (20)
7. A crank disk, such as is frequently found on horizontal engines of the overhanging crank type, is to be shrunk on and secured to the crank shaft. Explain how you would carry out the work, and how you would fit the key, showing particularly where the key would be bearing hard when finished. (20)

N.B.—*All Candidates are required to draw the slipper, Figs.* 5 *and* 6, *Plate B, in the way described.*

SECTION E.—SMITHS' WORK

Not more than four questions are to be attempted in addition to the drawing shown in Figs. 5 *and* 6, *Plate B.*
More marks are given for neat sketches than for vague and general descriptions.

1. Describe the forging of a small connecting rod, rectangular in section, and the small end of which would be suitable to work with the cross head shown in Plate B. (20)

9 1/4
pos reel